TRANSACTIONS

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS

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Meets: ***First Tuesday in Month***

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OF

American Society of Heating and Ventilating Engineers

No. 641

THE TWENTY-NINTH ANNUAL MEETING

1923

THE Twenty-ninth Annual Meeting of the Society attracted nearly 300 members and guests to Washington, D. C., where their reception and three-day stay were made memorable through the true Southern hospitality of the Washington members. More than 50 ladies were present whose time was pleasantly occupied in visiting historic spots in the city and vicinity and seeing how the government functions.

The meeting opened with a business session in the Engineering Societies Building, New York City, January 23, where the reports of the officers and committees were presented and the election of officers announced. The meeting then adjourned to Washington for the professional sessions, following the precedent established with such interesting results at Philadelphia in 1921.

The address of Herbert Hoover, Secretary of Commerce, paying a tribute to the work of the engineer in the daily affairs of the world and his valuable comment on the necessity of research work were outstanding features of the session at the Bureau of Standards, together with the inspection of various departments and the presentation and discussion of fundamental data offered by the Society's Research Laboratory.

Other matters of importance were the approval of the Standard Code Regulating the Installation of Warm Air Heating Furnaces in Residences, which had already been accepted by the *National Warm Air Heating and Ventilating Association, National Association of Sheet Metal Contractors,* and *Western Warm Air Furnace and Supply Association;* the approval of the new Constitution and By-Laws; the decision to continue an aggressive campaign for funds for the Research Laboratory; and a stirring appeal for the support of The Guide.

PROGRAM

FIRST SESSION

Tuesday, January 23, 10 a. m.

Engineering Societies Building, New York, N. Y.

Business Session:
- Announcement of Quorum
- Appointment of Tellers of Annual Election
- Address of President
- Report of Council
- Report of Secretary
- Report of Treasurer
- Report of Committees:
  - *a.* Executive
  - *b.* Finance
  - *c.* Publication
  - *d.* Membership
  - *e.* Revision of Constitution
- Unfinished Business
- Report of Tellers of Election
- New Business
- Adjournment of Session to Washington, D. C., for Professional Sessions

SECOND SESSION

Wednesday, January 24, 10 a. m.

Assembly Room—U. S. Bureau of Standards

Washington, D. C.

Professional Session:
- Address of Welcome
- President's Address
- Paper:
  - Heat Transmission of Building Structures, by E. F. Mueller
- Paper:
  - The Testing of Anemometers, by O. J. Hodge
- Paper:
  - Tests of Radiator Traps, by W. F. Stutz

THIRD SESSION

Wednesday, January 24, 8 p. m.

Convention Hall, Hotel Washington

Professional Session:
- Resume of Business Session at New York
- Reports of Technical Committees:
  - Committee on Chapters
  - Committee on Increase of Membership
  - Committee on Standard Code for Testing Fans
  - Guide Publication Committee
- Paper:
  - Heating and Ventilating U. S. Museum, by C. R. Denmark

Paper:
Heating and Ventilating the Bureau of Engraving and Printing, by N. S. Thompson
Paper:
Heating New Navy Gun Shop,
by M. S. Cooley

FOURTH SESSION

Thursday, January 25, 10 a. m.

Assembly Room, U. S. Bureau of Standards

Research Session:
Address of Hon. Herbert Hoover
Report of Committee on Research
The Research Laboratory, by F. Paul Anderson
Paper:
Capacities of Steam-Heating Risers as affected by Critical Velocity of Steam and Condensate Mixtures, by F. C. Houghton and L. Ebin

FIFTH SESSION

Thursday, January 25, 2:30 p. m.

Research Session (Continued):
Paper:
Some Physiological Reactions to High Temperatures and High Humidities, by Dr. W. J. McConnell and F. C. Houghten
Paper:
Determining Equal Comfort Lines,
by F. C. Houghten and C. P. Yagloglou
Paper:
New Data on Air Dust Determination,
by Margaret Ingels

SIXTH SESSION

Friday, January 26, 10 a. m.

Convention Hall, Hotel Washington

Heating Session:
Paper:
Some Comments on Warm Air Furnace Installation,
by F. T. Giblin and Frank K. Chew
Report of Committee on Code for Installation of Warm Air Furnaces
Talk on English Heating Practice,
by W. Nelson Haden
Paper:
Humidity Requirements for Residences,
by A. P. Kratz

SEVENTH SESSION

Friday, January 26, 2 p. m.

Convention Hall, Hotel Washington

Ventilation Session:
Paper:
Modern Tendencies of Ventilation Practice,
by Dr. George T. Palmer

Paper: Report of New York Commission on Ventilation

Comparative Tests of Roof Ventilators, by Prof. F. B. Rowley

REPORT OF THE COMMITTEE ON RESEARCH

BEFORE, in any way, going into the details of the conduct of the Society's Research Laboratory, it seems most fitting that we briefly review the thought, about research in general, among men of affairs in all walks of life, as expressed during the past year.

The increasing recognition of the need for research is positively amazing. That research is an important factor in our civilization and future progress as a nation is an idea repeatedly expressed and enlarged upon by the daily press, magazines, national engineering societies and other scientific bodies.

I do not think I could express this idea any better than by quoting from a recent editorial appearing in one of our great daily newspapers:

> Guesswork is the chief curse of business today. We have reached a time when laboratory methods must be applied to commercial phenomena. The moment has come when this Nation must take up scientific research whole-heartedly or relinquish world leadership.
>
> The field of research is not only a virgin land of romance and adventure, but it is a sphere of activity that will net surprising profits to those who engage in it. Some of our great industries have recognized this fact and are undertaking co-operative scientific investigation in systematic fashion on a national scale. The tanners took up research less than two years ago, and the solution of one problem will save this great American business two per cent on its total production.
>
> The moral of the story is that valuable benefits always accrue in the conduct of business when facts replace guesswork. It is costly in industry to base important activities on nothing more substantial than mere conjecture. A vigorous campaign of national research would lower materially the cost of everyday necessities. The progress and safety of America depend on mentality more than on materials. Research is a first-line national defense.

The prosecution, therefore, of research work we can take for granted as being not only patriotic and desirable from an idealistic standpoint, but also profitable when viewed from both the intellectual and the materialistic standpoints.

We, as a national engineering society, can, I think, with perfect modesty, congratulate ourselves that in originating and carrying on our Research Laboratory, real pioneering is being done.

To those of us in close contact with the work of the Laboratory there have been many encouraging moments, for recognition of its worth is

rapidly becoming universal. We are extremely fortunate, for history tells us many instances where profound and far-reaching advances in science and civilization have been extremely slow of acceptance. One need only to mention the names of Roger Bacon, Columbus, Copernicus, Galileo and recently our own Professor Langley to realize that this is true.

At the beginning of the year just passed, a policy was inaugurated of bringing the Laboratory more closely in touch with its subscribers, those interested in the field of heating and ventilating, and the members of our Society. The methods used to accomplish this were:

1. Your Chairman wrote to every subscriber and accompanied this with an up-to-date pamphlet telling of the work being done and asking for suggestions.
2. Our Director wrote frequent personal letters to a large number of the subscribers about particular problems, and sent progress bulletins to others.
3. A monthly communication regarding the Laboratory was published in the Society's JOURNAL.
4. Director Anderson, accompanied by your Chairman, made addresses to the Cleveland, Pittsburgh, Western New York and Philadelphia Chapters, to the United Engineering Society in Schenectady, N. Y., and Director Anderson, in addition, addressed the New York, Toronto, Michigan and Illinois Chapters, and several other scientific bodies.

Co-operation with all engineering, scientific and educational bodies has been encouraged and stimulated wherever time and funds have permitted. A definite agreement covering co-operation with the *American Institute of Architects* has been entered into, and the measurement of air leakage around windows is the first problem scheduled. The *American Institute of Architects* is contributing the necessary funds for this work.

Other organizations with whom we are co-operating are Pennsylvania State College, National Research Council, *Civilization and Climate Association,* whose head is Dr. Ellsworth Huntington, as well as a number of others mentioned in previous reports.

The Johnson Service Co. have generously offered us the use of their new heat research laboratory for any investigative work for which we can utilize it.

The splendid co-operation which we have received from the various officials of the Bureau of Mines and all members of its staff since the inception of our research work has generously and whole-heartedly continued throughout the past year.

Our staff of investigators at Pittsburgh has been increased and now we have engaged on our work in addition to our Director three (3) Research Heads and five (5) Research Assistants.

We are, therefore, able to present at this meeting papers covering three separate finished research investigations. At last two of these are of startling importance to all mankind. Never before in the whole history of civilization, so far as can be ascertained, has any conclusive

reliable data of this character been given to the scientific world. A milestone in man's progress in scientific investigation has been passed. At last the relation of man to his atmospheric environment is not altogether a closed book

From the work conducted in our constant temperature rooms we present to you in finished form—*first*—A chart on which are drawn equal comfort lines or lines which tell what combinations of temperature and humidity give the same degree of comfort and physiological reactions, and *second*—The determination of the actual physiological reactions of human beings at rest in still air containing high humidities.

In addition to these two far-reaching conclusive reports, we present a finished, practical, simple yet scientific instrument for the measurement of the dust in the atmosphere. There have been devised fifty or more dust measuring instruments, most of which required a scientist of high degree for accurate observation, but our Laboratory has been able in a relatively short space of time to perfect an instrument which is not only more accurate and reliable than others previously developed, but which is also so simple that any engineer may operate it with success.

By far the knottiest problem which confronted your Committee at the beginning of the year just passed was the securing of additional funds, and we have not yet solved the problem. No active campaign for funds was organized and carried on in 1921 on account of the general business depression which existed.

In 1922 there was not only the condition of the decreased available subscriptions, but also the increased and increasing expenses of the Laboratory to be considered. Your Committee, therefore, decided that, as prosperous conditions were not at all general at the beginning of 1922, it would be best to set a modest quota and strive diligently to secure it.

Last year our budget—August, 1921 to August, 1922—was $27,710.00, and expenses $27,664.60, but our paid-in subscriptions amounted to only $22,594.81.

This year our budget is $27,500.00, and our paid-in subscriptions to date only $22,594.05.

Respectfully submitted,

COMMITTEE ON RESEARCH,

THORNTON LEWIS, *Chairman*.

ADDRESS OF DIRECTOR F. PAUL ANDERSON ON THE RESEARCH LABORATORY

THIS Research Laboratory, being conducted by the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, in my opinion, has accomplished something.

When you stop to contemplate what the engineering societies of this country have done you must feel a just pride in realizing that your little Society has taken a step in advance of any other scientific society in

America. Perhaps your ideals and your dreams have been too exalted, but nevertheless the fact stands out that you have undertaken something that has never been projected by any other organization of engineers in America.

The co-operation that has been received by this Laboratory at all hands is in my opinion very remarkable. On every side there has been a sympathetic interest in the work that has gone on. The Bureau of Mines, where this Laboratory happens to be stationed, has lent every possible aid to your Society. I do not know just how it has been accomplished, but I attribute it very largely to those wise and splendid men who conceived the idea of a Laboratory and placed it in this environment, because there is every indication that there is a desire to get away from the set routine of the day, as far as the great institution of the Experiment Station of the U. S. Bureau of Mines is concerned, and help this Laboratory of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS.

Everything is given to us that we want; mathematicians, photographers, computers, investigators of all sorts, instrument makers, skilled mechanicians—in fact, anything that we ask for, appears as if produced by the magic of Aladdin's lamp. We have at our command there, perhaps, at a very conservative estimate, 25 men who can be called upon to serve this Society. The instrument that was presented to you today was made in the shops of the Bureau of Mines. I was interested in the desire on the part of the instrument makers to produce something that would stand, perhaps, as a permanent instrument in the scientific field. I will say that no laboratory has ever existed in a more enchanted atmosphere than does our little Laboratory established in connection with the Experiment Station of the U. S. Bureau of Mines.

Today I am going to call your attention to the high spots of our achievements in research because the papers themselves will tell you more definitely all of the story, and these expositions will be told by those who have been responsible for the development of the particular studies, that will be presented to you

There will be presented today a few papers that I am sure will give to every person present a thrill as the import of each is realized. Two of our investigations are in a way conclusive. First is the development of a heat meter by Mr. Nicholls. He has developed an instrument which we think will measure definitely how many heat units pass through a substance under variable differences of temperature conditions.

I have heard engineers for years talk about comfort lines. I think I can impersonate Mr. Carrier, not absolutely but in essential, when I tell you that he expressed to me the yearning that if we could determine the comfort lines which, like critical velocity, is a misnomer, and I agree entirely with the idea that was expressed here that we should have definitions, that if our psychrometric rooms could be equipped and we could determine once for all the comfort lines of equivalent temperatures, then we could afford to close up the Society's Laboratory and consider the money already expended well spent. Mr. Carrier, I am sure did not mean specifically that he would be willing to see the Laboratory closed

up, but his thought was that if that particular investigation could be brought to a conclusion and there could be given to the engineering fraternity comfort lines, then he would be willing to say that this Society has brought to a realization all it could ever have hoped to accomplish. Gentlemen, those comfort lines have been determined by Mr. F. C. Houghten beyond any question, and he is presenting them to you today. They are monumental, in my opinion, in the annals of science.

Another conclusion that we are very glad to record is the effect of high temperatures and of high humidities on physiological reactions. We have heard a great deal said about what is the extreme temperature in which man can live at rest. Nobody knows that; everybody will know now. But through the co-operation of the United States Public Health Service and the engineers of your Laboratory there has been determined a positive limit of combined dry and wet bulb temperatures at which human beings can exist at rest over prolonged periods. In other words, we are very sure that we can say to you now that 112 deg. fahr. dry bulb and 30 per cent relative humidity represents the extreme limit under which man can exist for any considerable time with safety.

Now it was a very interesting thing in getting action of this kind through co-operation. Dr. R. R. Sayers, I think, is responsible very largely for this scientific triumph. Dr. Sayers was interested in the problem. He took the possibilities of the problem before his chiefs and he was able to induce them to send a number of high class men who have subjected themselves to exacting ordeals, even going beyond the limit of endurance until they had at times to be taken out of the room, and for days afterwards they were in an abnormal physical condition. These sacrificing persons went beyond nature's limit in order to determine a physical law of value.

There are four elements to this particular problem and we are going to give you one of them today. The solution of these four conditions represent perhaps two years' work. Dr. Sayers and I have discussed this matter and we have come to the conclusion that it would take two years to complete the work if the same degree of speed is maintained that we have shown up to the present time.

Another paper to be presented today describes an instrument which can be carried into any industry and be used effectively to determine the amount of dust in the atmosphere.

Now, gentlemen, whether these facts will thrill you with pride I do not know, but they do me. I can tell about them because I am not responsible for them. I simply happened to have been associated with a lot of scientists who have achieved and therefore I can say as much about all these results as I like.

I could talk to you for some time in reference to my enthusiasm and the possibilities of this Laboratory and what can be accomplished. I think of this Research Laboratory as a child of the Society, and he is worth thinking about.

The Laboratory is resting on the firm foundation of service and every one of the staff is happy in the process of adding to the recorded scientific truths and contributing to the health and comfort of his fellow beings.

No. 642

REPORT OF COMMITTEE ON REVISION OF CONSTITUTION

AFTER several meetings of the Committee on Revision of Constitution, changes in the Constitution have been agreed upon, and the resulting draft is submitted to the members of the Society, who are requested to study it and prepare their comments for presentation at the Annual Meeting, when this subject comes up for action by the Society. The Constitution in this form will be submitted to the Annual Meeting, and if there approved will be voted upon by mail ballot according to the method of amendment laid down by the present Constitution.

It is being presented well in advance for the convenience of the members.

COMMITTEE ON REVISION OF CONSTITUTION,
CHAMPLAIN L. RILEY, *Chairman.*

CONSTITUTION AND BY-LAWS

CONSTITUTION

ARTICLE I.

NAME AND OBJECT

Section 1. The name of this Society shall be the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS.

Section 2. The Society is organized to advance the arts and sciences of heating and ventilation by research, discussion, intercourse and publication.

ARTICLE II.

MEMBERSHIP

Section 1. Persons connected with the arts and sciences related to heating and ventilation may be eligible for admission into the Society.

Section 2. The membership of the Society shall consist of Honorary Members, Members, Junior Members, Associate Members and Student Members.

Section 3. An Honorary Member shall be a person of acknowledged professional eminence.

Section 4. A Member shall be twenty-eight (28) years of age or over and shall be a person of experience in the arts or sciences of heating and ventilation, and shall have been in active practice of his profession and in responsible charge of important work for five (5) years and shall be qualified to design as well as to direct such engineering work. Fulfilling the duties of a professor of heating and ventilating engineering in a college or technical school of accepted standing shall be taken as an equivalent to an equal number of years of active practice. Graduation from a school of engineering of recognized standing shall be considered as equivalent to two (2) years of active practice.

Mining, civil, electrical, mechanical, naval or government engineers, chemists, physicians, scientists or architects, who are qualified by reason of their experience in designing, improving, inspecting, investigating or developing the arts or sciences of heating and ventilation, are also eligible to membership.

Section 5. A Junior Member shall be a person over twenty (20) years and under twenty-eight (28) years of age, who has been actively engaged in the work of heating and ventilation for three (3) years, or is a graduate of a school of engineering of recognized standing.

Section 6. An Associate Member shall be twenty-five (25) years of age or over. He need not be an engineer, but must have been so connected with some branch of engineering or the arts or sciences of heating and ventilation or the industries relating thereto, that he may be considered as qualified to co-operate with heating and ventilating engineers in the advancement of profssional knowledge.

Section 7. A Student Member shall be a person under twenty (20) years of age who has had sufficient experience to qualify for a higher grade of membership.

Section 8. All grades of membership shall enjoy equal privileges in the Society, excepting as follows:

Junior Members and Student Members shall have no vote nor hold office in the Society.

ARTICLE III.

ADMISSION

Section 1. Honorary Members shall be nominated by at least ten (10) members of the Society. The grounds upon which the nomination is made shall be presented in writing and be signed by the ten (10) proposers.

Section 2. Election to any grade of membership in the Society, except Honorary Membership shall be by an affirmative vote of two-thirds of the Council.

Section 3. When a Student or Junior Member reaches the age limit of his grade, his membership shall automatically terminate.

ARTICLE IV.

THE COUNCIL

Section 1. The affairs of the Society shall be managed by a Board of Directors chosen from among the persons entitled to vote, which shall be styled "The Council." It shall consist of twelve (12) members, four of whom are the duly elected officers. The Secretary may take part in the deliberations of the Council, but shall have no vote therein.

Section 2. The Council shall regulate its own proceedings and may delegate specific powers to any Committee or to any one or more members by resolution.

Section 3. Should a vacancy occur in the Council, or any elective office, through death, resignation or other cause, the Council may elect a Member to fill the vacancy, pending the next annual election.

Section 4. The members of the Council, or Directors shall be held harmless by the Society from any liability to third persons, resulting from any acts in their capacity as such Directors or while engaged in the affairs or business of the Society, unless caused by the wilful misconduct of any such Director or Directors, which liability the Society hereby assumes.

Section 5. No act of any Committee or any delegate shall be binding until it has been approved by a resolution of the Council.

Section 6. An act of the Council, which shall have received the expressed or implied sanction of the membership at the next subsequent meeting of the Society shall be deemed to be an act of the Society, and shall not afterwards be impeached by any member.

ARTICLE V.

OFFICERS

Section 1. At each Annual Meeting there shall be elected from among the members the following officers:

a. President.

b. First Vice-President.
c. Second Vice-President.
d. Treasurer.
e. Seven members of the Council, all of whom shall hold office for one year

Section 2. The term of all elective officers and members of the Council shall begin on the adjournment of the Annual Meeting of the Society. An officer shall continue in office until his successor has been elected and has accepted office.

Section 3. The Council at its first meeting after the Annual Meeting shall appoint a Secretary of the Society for one year. The Secretary shall be subject to removal for cause by affirmative vote of the majority of the members of the Council.

Section 4. The President, Secretary and Treasurer shall perform the duties usually pertaining to their respective offices and such other duties as may be provided for in the By-Laws or required of them by the Council.

ARTICLE VI.

MEETINGS

Section 1. The Society shall hold two meetings in each year. The Annual Meeting shall begin in the City of New York, N. Y., on the fourth Tuesday in January, continuing from day to day as the Council may arrange.

Section 2. The Semi-Annual Meeting shall be held at such time and place as the Council may elect.

Section 3. Special Meetings of the Society may be called at any time at the discretion of the Council, or shall be called by the President upon written request of ten (10) members in good standing, the notices for such meetings to state the business for which such meeting is called and no other business shall be entertained or transacted at that meeting.

ARTICLE VII.

AMENDMENTS

Section 1. At any meeting of the Society any person entitled to vote may propose in writing an amendment to this Constitution. Such proposed amendment shall not be voted upon at that meeting, but shall be open to discussion and to such modification as may be accepted by the proposer. The proposed amendment shall be mailed in printed form by the Secretary to each member entitled to vote, at least sixty (60) days previous to the next Annual Meeting, accompanied by comment by the Council, if it so elects. At that Annual Meeting such proposed amendment shall be presented for discussion and final amendment. It shall subsequently be submitted to all members entitled to vote, provided that one-third of those present, which in number shall not be less than fifteen (15) vote in favor of such submission. The final vote on adoption shall be by sealed letter-ballot, closing at 12 o'clock on the first Monday of April following the Annual Meeting.

Section 2. A letter-ballot, accompanied by the text of the proposed amendment, shall be mailed by the Secretary to each member entitled to vote, at least thirty (30) days previous to the closure of the voting. The ballots shall be voted, canvassed and announced as provided in the By-Laws. An amendment receiving a majority of the votes cast shall be adopted. An amendment shall take effect on the announcement of its adoption by the presiding officer at the Semi-Annual Meeting next following the closure of the vote.

Section 3. Any changes in the order of consecutive numbering of existing articles or sections of the Constitution, made necessary by such adopted amendment, shall be under the direction of the Council.

BY-LAWS

ARTICLE I.

HEADQUARTERS

Section 1. The headquarters of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS shall be located in the City of New York, N. Y., at such address as the Council may select.

ARTICLE II.

Section 1. A candidate for admission to any grade of membership in the Society, except Honorary Membership, must make application on a form approved by the Council, upon which he shall write a statement giving a complete account of his qualifications and engineering experience, and an agreement that he will, if elected, conform to the Constitution, By-Laws and Rules of the Society.

Section 2. A candidate for any grade of membership in the Society, except Honorary Membership, must be proposed by two members, to whom he must be personally known, and such application must be seconded by two other members.

Section 3. The proposers of a candidate may be required to submit a complete account of the qualifications of the candidate, including a statement in writing of the professional experience and the character and diversity of the work performed, age, education and any other qualifications the Council may demand.

Section 4. In case applicants for membership are not acquainted with members of the Society, endorsements from the faculty of educational institutions, or the recommendations of the members of other professional societies, or by the officers of responsible corporations, who are well acquainted with the applicant, may be considered. If sufficient evidence has been submitted to satisfy the Membership Committee that the applicant is worthy of admission to membership.

Section 5. Honorary membership shall be proposed at an Annual Meeting only and all such proposals shall have the unanimous endorsement of the Council before they shall be submitted to the meeting for election.

Section 6. All applications for membership are to be sent to the Secretary and shall be acted upon by the Membership Committee as soon as possible. The names of the candidates passed upon favorably by the Membership Committee shall be printed in the next issue of the Journal or sent to the members in other approved manner as ordered by the Council, accompanied by a notice inviting the members of the Society to submit in confidence information regarding the qualifications of the candidates.

Section 7. Following the publication of the names of candidates in the Journal or other publication as may be provided, and in no case less than thirty days after such publication, the Council shall vote upon the election of the candidates proposed for membership.

Section 8. If two-thirds of the Council shall cast a vote favorable to a candidate he shall be elected, and his name shall be published in the next issue of the Journal.

Section 9. The names of the candidates who are not elected shall neither be recorded nor announced in the proceedings of the Council.

Section 10. Any person having been elected to membership in the Society shall be promptly notified by the Secretary, whereupon he shall accept such election, subscribe to the Constitution, By-Laws and Rules of the Society, and pay the initiation fee within three months after such notice of election shall have been sent him, or his election shall become void.

ARTICLE III.

INITIATION FEES AND DUES

Section 1. The initiation fee of Members and Associate Members shall be fifteen dollars ($15.00); of Junior Members ten dollars ($10.00), and of Student Members five dollars ($5.00).

Section 2. The annual dues of Members and Associate Members shall be fifteen dollars ($15.00); of Junior Members twelve dollars ($12.00) and of Student Members five dollars ($5.00).

Section 3. Of the annual dues paid by the members of each grade, three dollars ($3.00) shall be considered as a subscription paid for the JOURNAL.

Section 4. All dues shall be payable in January of each year in advance. The dues of new members of all grades shall be due and payable on the first day of the month following the date of admission of such members.

Section 5. The dues of a new member of any grade except a Junior Member may be pro-rated monthly for the balance of the year, but if the amount thus paid is less than five dollars ($5.00) such member shall not be entitled to receive the volume of the Transactions for the year in which he is elected, but he shall otherwise be entitled to all the rights and privileges of membership.

Section 6. Junior Membership shall be limited to eight (8) years and upon election to a higher grade such members shall, upon notification of transfer, pay an additional initiation fee of five dollars ($5.00) and thereafter pay the annual dues of the grade to which they are transferred.

Section 7. A member of any grade who pays five dollars ($5.00) or less dues in any year shall not be entitled to receive the Transactions of the Society, but may purchase the same at members' rates; otherwise he shall be entitled to all the rights and privileges of his grade of membership.

ARTICLE IV.

RESIGNATIONS AND EXPULSIONS

Section 1. Any member whose dues are paid in full may resign at any time. Resignations must be presented in writing to the Council and action must be taken at the first meeting of the Council following receipt of same.

Section 2. Any member whose dues shall remain unpaid for one year shall forfeit the privileges of membership, and if he neglects or refuses to pay his dues within thirty days after notification from the Secretary, his name shall be stricken from the roll of members.

Section 3. The Council, by a two-thirds majority vote, may expel any member of any grade who may be adjudged to have violated the Constitution or By-Laws of the Society, or who shall have been found guilty, after an adequate opportunity of a hearing, of conduct rendering him unfit to continue in its membership.

Section 4. Any person ceasing to be a member of the Society, through resignation or otherwise, shall forfeit all right, title and interest in the property of the Society.

Section 5. Any person who has been a member of the Society for fifteen (15) years or more and has retired from business, then upon reaching the age of seventy (70) years, shall have his dues remitted for the current year and for ensuing years, without surrendering any of the privileges of membership as long as he lives.

ARTICLE V.

MEETINGS

Section 1. Professional Meetings of the Society for the reading and discussion of papers and for topical discussions may be held at such times and places as the Council may direct.

Section 2. Announcements of all meetings of the Society shall be published in the JOURNAL and the Secretary shall also mail a notice of same to each member of all grades, not less than thirty (30) days before the date of each meeting.

Section 3. A quorum for the transaction of business at any meeting of the members shall consist of not less than fifteen (15). Each member shall be entitled to one vote only on the question.

Section 4. The order of business at the Annual Meeting of the Society shall be as follows:

- *a.* Roll call;
- *b.* Minutes of previous Meeting;
- *c.* Report of Officers;
- *d.* Report of Council;
- *e.* Reports of Committees;
- *f.* Report of Tellers of Annual Election;
- *g.* Unfinished Business;
- *h.* New Business;
- *i.* Reading and Discussion of Papers;

j. Installation of Officers;
k. Adjourn.

Section 5. All questions arising at any meeting involving parliamentary rules not provided for in these By-Laws, shall be governed by "Robert's Rules of Order."

ARTICLE VI.

THE COUNCIL

Section 1. Five members of the Council shall constitute a quorum for the transaction of business at any Council Meeting.

Section 2. The Council shall hold meetings, have an office and keep the books of the Society at the office selected as headquarters in the City of New York, N. Y. Meetings of the Council may at its convenience be held at such other place or places as the Council from time to time may determine.

Section 3. Regular meetings of the Council shall be held quarterly.

Section 4. Special Meetings of the Council shall be held whenever called by the President or by not fewer than three (3) Council members.

Section 5. The Order of Business at any Regular Meeting of the Council shall be as follows:

a. Minutes of previous Meeting;
b. Report of Committees;
c. Report of Officers;
d. Motions and Resolutions;
e. Miscellaneous Business.

Section 6. The Council shall present at the Annual Meeting a complete report of the work of the Society during the preceding year, together with a record of the attendance of the members of the Council.

Section 7. The Council shall designate the bank or depository in which the funds of the Society shall be deposited and shall by appropriate resolutions designate the purpose for which the funds may be withdrawn and authorize such withdrawal.

ARTICLE VII.

OFFICERS, POWERS AND DUTIES

Section 1. The President shall be the chief executive officer of the Society. He shall preside at all meetings of the Society and of the Council. He shall have general charge and supervision of the business and affairs of the Society. He shall sign in behalf of and in the name of the Society, all contracts and all certificates of membership in the Society. He shall make a report of the affairs of the Society to the Annual Meeting and shall do and perform such other duties as usually appertain to his office. He shall not be eligible for immediate re-election to the same office at the expiration of the term for which he was elected.

Section 2. The First Vice-President shall possess all the powers and perform the duties of the President in his absence or disability. He shall perform such other duties as may be assigned to him by the Council.

Section 3. The Second Vice-President shall posesss all the powers and perform the duties of the President in the absence or disability of the President and First Vice-President. He shall have such other powers and perform such other duties as may be assigned to him by the Council.

Section 4. The Treasurer shall have the custody of all the funds of the Society, and shall deposit them to the credit of the Society in such bank or depository as the Council may designate, and he shall disburse the same as provided in Art. VIII, Sec. 3. He shall enter regularly in books of the Society to be kept by him for the purpose, full and accurate account of all moneys received and disbursed for the Society. He shall at all reasonable times exhibit his books and accounts to any member of the Council, and shall perform all duties incident to the office of Treasurer, subject to the control of the Council. He shall give a bond in a penal sum and with a surety or sureties approved by the Council, for the faithful per-

formance of his duties as Treasurer. If a surety company bond is furnished the premiums therefor shall be paid by the Society.

Section 5. The accounts of the Treasurer and the books of the Society shall be audited annually by a certified public accountant selected by the Council at least thirty (30) days before the close of the fiscal year.

Section 6. The Secretary shall keep the minutes of all meetings of the Council and the minutes of all meetings of the Society. He shall attend to the giving and serving of all notices of the Society and of the Council. He shall sign with the President in the name of the Society all contracts or official documents requiring the corporate signature and shall affix the seal of the Society to such papers or documents as require the same. He shall have charge of such books and papers of the Society as the Council may direct, all of which at all reasonable times shall be open to the examination of any member of the Society. He shall in general perform all the duties incident to the office of Secretary and shall perform such other duties as may be assigned to him by the Council. He shall give a bond in a penal sum and with a surety or sureties approved by the Council, for the faithful performance of his duties as Secretary. If a surety company bond is furnished the premium therefor shall be paid by the Society. Assistant secretaries may be appointed by the Council.

Section 7. The Secretary shall receive a salary which shall be fixed by the Council. No other officer of the Society shall, as such, receive any salary or compensation for his services.

ARTICLE VIII.

COUNCIL COMMITTEES

Section 1. At the first meeting of the Council after the Annual Meeting, the President shall appoint from the members of the Council the following committees, consisting of three (3) members each, to act under the direction of the Council:

a. Executive Committee;
b. Finance Committee;
c. Membership Committee;
d. Publication Committee;
e. Research Committee.

Section 2. The Executive Committee shall investigate and make recommendations to the Council regarding all matters having to do with the professional or business status of the Society and matters which might in any way reflect upon one or more members of the Society, or the Society as a Whole. It shall exercise such other functions as the Council may specify.

Section 3. The Finance Committee shall prepare an annual budget of expenditures for the Society and shall pass upon and approve in writing all expenditures authorized by the Council. No expenditures are to be made by the Secretary, except for salaries, traveling expenses and petty cash, unless authorized by the Chairman of the Finance Committee, on forms provided for that purpose.

In case of disability or absence of the Treasurer the Chairman of the Finance Committee is authorized to sign checks. He shall give bond in the same manner as provided for the Treasurer.

Section 4. The Membership Committee shall receive from the Secretary all applications for membership, make rigid inquiry as to the eligibility of candidates, and report to the Council only such as have been approved. In case of disapproval, only the proposers of the applicant shall be notified of such action. The proceedings of the Committee shall be private and confidential.

Section 5. The Publication Committee shall receive and examine all papers for presentation to the Society and accept such as it may approve. This Committee shall review the papers and discussions which have been presented at the meetings, and shall decide what papers and discussions, or parts of the same, shall be published. It shall publish the Transactions of the Society, containing the papers and discussions so approved, and abstracts of the minutes of the Society and of the Council. No member shall publish any papers as having been read before the Society without obtaining the consent of this Committee, and such consent shall

not be construed to be an endorsement by the Society of any statements contained in such papers or publications.

ARTICLE IX.

ELECTIVE COMMITTEES

Section 1. The Research Executive Committee, as provided for under the regulations creating the Research Laboratory, shall prepare the budget for the expenditures of the Research Laboratory for the current year, and shall pass upon and approve in writing all expenditures authorized by the Research Committee. The payment of bills shall be authorized in the same manner as provided in Article VIII, Section 3, excepting that the Chairman of the Research Executive Committee shall act instead of the Chairman of the Finance Committee, and the Director of the Research Laboratory shall act instead of the Secretary of the Society, but in case of absence or disability on the part of the Treasurer the checks may be signed by the Chairman of the Research Committee, instead of the Chairman of the Finance Committee.

Section 2. A Nominating Committee consisting of five (5) members of the Society not office holders, shall be elected at the Annual Meeting, to select one or more candidates for each of the offices to be filled at the next Annual Meeting. This Committee shall present to the Secretary, at least four (4) months before the Annual Meeting, the names of the candidates for the offices to be filled, having first obtained the consent of the members selected. The Secretary shall publish the names of such candidates in the October issue of the JOURNAL.

Section 3. The Council shall have power to delegate to any other committee any duties that may be deemed necessary or expedient in the interests of the Society and such committee shall report to the Council when and as required. The Council shall also appoint any committee and direct its proceedings as may be ordered by resolution of the Society at any meeting.

ARTICLE X.

ELECTION OF OFFICERS

Section 1. Nominations for Officers and Members of the Council may be made by the Nominating Committee.

Section 2. Ten or more members of the Society may present to the Secretary, over their signatures, the name of any member of the Society as a candidate for any office, provided such name is presented at least sixty (60) days preceding the next Annual Meeting, and the Secretary shall add such names to the ballot if they are not already included in the list of names presented in the formal report of the Nominating Committee. Such names when presented shall be included on the printed ballot, with special notation that they are presented by members independent of the Nominating Committee report.

Section 3. The Secretary shall prepare ballots with the names of all candidates and forward them to the members, at least thirty (30) days before the date of the Annual Meeting.

Section 4. Each member entitled to vote shall cancel the names of all candidates for whom he does not wish to vote and return his ballot so that it will reach the Secretary before 9 o'clock in the forenoon of the day of the Annual Meeting. A member may write upon his ballot the name of any member for whom he wishes to vote, if such name is not on the printed ballot. The ballot is to be enclosed in a blank envelope which shall in turn be enclosed in another one endorsed by the voter.

Section 5. The Secretary shall prepare a list of the members who have not paid their dues within the year preceding the Annual Meeting, and no vote shall be counted for any member whose name is endorsed on the outer envelope if such name shall appear in the list of delinquents.

Section 6. The ballots shall be opened, and the result of the vote declared on the first day of the Annual Meeting by three tellers appointed by the President.

The candidates receiving the most votes for the several offices shall be declared elected, and shall take office at the close of the last session of the Annual Meeting.

Section 7. In the event of a tie vote at any election of officers of the Society, the Council, by a majority vote, shall decide the tie.

ARTICLE XI.

LOCAL CHAPTERS

Section 1. Local Chapters of the Society may be formed upon application of ten (10) members if the organization of such local Chapter will, in the judgment of the Council of this Society, advance the Society's interests.

Upon recommendation of the Council, a charter may be granted by the Society to form such local Chapter, which shall be operated and conducted under the control and at the pleasure of this Society; such local Chapter shall be governed by the Constitution and By-Laws of this Society and by such other local By-Laws as may be adopted by the local Chapter and approved by the Council of this Society before becoming operative.

The membership of such Chapter shall comprise only members of the different grades in good standing in this Society; any member of any local Chapter who shall cease to be a member of this Society shall thereby forfeit all right to membership in such local Chapter.

Every such local Chapter when formed shall be chartered in the name of the State, section of State, County or City, in which the same shall be located.

Section 2. Each Chapter shall appoint a Publication Committee, which shall read all papers presented at any meeting of the Chapter, or if an address is made without notes, the Committee shall listen to the address and decide whether or not it shall be recommended for publication, to the Publication Committee of the Society, and the decision of the latter shall be binding upon all members.

Section 3. Any person delivering an address before any Chapter of this Society shall be asked not to give it out for publication without first presenting it for the approval of the Publication Committee of the Society.

Section 4. The charter of a local Chapter may be revoked, for cause, by majority vote of Council.

ARTICLE XII.

SOCIETY'S ENDORSEMENT

Section 1.. Recommendation, indorsement or approval by the Society or the Council shall not be given to or made for any individual, partnership, corporation or association, nor of or for any scientific, literary, mechanical or engineering production, other than as affecting public welfare, and when affecting public welfare the opinion so expressed shall not convey any intent to promote the interests of any individual, partnership, corporation or association, nor shall such expression be harmful to any other similar interests, nor shall it differentiate between any two similar methods, processes, devices or apparatus by special commendation of one over the other.

The Society may, however, receive papers and reports on experiments, improvements and developments of every character affecting the arts and sciences of heating and ventilation and it may freely discuss same and have the proceedings published in the Transactions of the Society.

Section 2. The work conducted by the Research Laboratory of this Society shall be confined to a determination of the basic or fundamental principles and laws underlying all matters in the field of heating and ventilation relating to the health and comfort of the human system. As the work progresses and reports are made to the Society such reports are not to be construed as an indorsement or repudiation by the Research Laboratory or by the Society, unless a specific statement of indorsement or repudiation is included.

ARTICLE XIII.

AMENDMENTS

Section 1. The By-Laws of the Society shall be subject to addition, amendment or repeal by a majority of the members present and voting at any Annual Meet-

ing of the Society, or at any Special Meeting of the members called for that purpose, provided that a copy of such proposed addition, amendment or repeal shall have been submitted in writing to all of the members entitled to vote, at least thirty (30) days prior to the time of the meeting at which such addition, amendment or repeal is to be considered.

No. 643

MODERN TENDENCIES OF VENTILATION PRACTICE

By George T. Palmer[1], Dr. Ph., Detroit, Mich.

Member

"VITIATED air produces deformity, imbecility and idiocy." Thus wrote an American physician in 1850. And that is not all that vitiated air was charged with. It encourages "Pusillanimity and cowardice," "vice," "intemperance in the use of intoxicating drinks." It "produces inaptitude for study and, therefore, ignorance."

These contentions sound a bit exaggerated. Surely the most rabid open air enthusiasts, even those intrepid individuals who break the ice for a stimulating dip in midwinter, would hardly dare to be so dogmatic today.

In fifty years' time we have learned to be analytical, to separate a problem into its elements, to consider a number of possibilities before drawing conclusions. This has been made possible by an advancing knowledge of physics, chemistry, physiology and sociology. Bad ventilation is detrimental to health. This has been proved. That ventilation is at the base of all our social shortcomings is a stupendous exaggeration. It is not only untrue, but it is unnecessary to charge so many evils to bad ventilation. There are reasons scientifically sound for demanding good ventilation on the grounds of comfort, health and efficiency without going further.

Perhaps the most outstanding change in ventilation since 1870 has been the substitution of experimentation for guess-work. It is for this reason that we are vastly nearer the true causes of bad ventilation today, and we are in consequence in a better position to prescribe effective remedies for bad ventilation.

CONCEPTIONS OF VENTILATION HAVE CHANGED

Ventilation, as we understand it today, has outgrown its etymological swaddling clothes. Literally, ventilation means "to air", or by fanning or blowing "to replace foul air with pure air." This definition is some-

[1] Epidemiologist, Department of Health.

Paper presented at the Annual Meeting of the American Society of Heating and Ventilating Engineers, Washington, D. C., January, 1923.

what narrow and restricted now because in its broad meaning ventilation involves certain physical characteristics of air, such as temperature, humidity and movement, factors which the words "foulness" and "purity" do not suggest. Broadly, ventilation is the adjustment of the indoor air environment to meet the requirements of comfort, health and efficiency.

Our conceptions of the causes of bad ventilation have, it is true, changed in 50 years, but unchanged still is the belief in the necessity for ventilation. Today in many offices, schools, factories and theatres, we can experience that sensation of stuffiness, oppression, discomfort, uneasiness, which are the products of poor ventilation. These same uncomfortable feelings were well known in the 70's. They were known and described long before that time. All mankind is and has been conscious of the fact that you cannot crowd human beings into an enclosure and expect them to remain in comfort unless some provision is made for an interchange of or an alteration in the air.

THE REASONS FOR VENTILATING

Human beings give off heat, moisture and odor. Through respiration they convert oxygen into carbon dioxide. Our ordinary habitations are not air-tight, and there is constantly taking place an interchange of air between the inside and outside, through walls and through cracks around doors and windows. This natural interchange of air is usually sufficient to maintain comfort when there is ample space and there are few people. As the ratio of people to space is increased the natural interchange of air becomes increasingly less effective in keeping down the accumulation of heat, moisture and odor. The crowding may proceed much further, however, before oxygen is diminished or carbon dioxide is increased to the point of discomfort or danger. In short, the body's ability to withstand alterations in the chemical substances, oxygen and carbon dioxide, is vastly greater than its ability to withstand changes in the physical characteristics of the atmosphere. Furthermore, the natural diffusion of the chemical constituents is much more rapid than the diffusion of heat and odor.

The internal body temperature in health is approximately constant, being maintained at all seasons of the year within one or two degrees either side of 98 deg. fahr. Although man can adjust himself to a wide range of external air temperature, there is an optimum in the neighborhood of 65 deg. fahr. This is attested to by the history of civilization, for the greatest advances have been made in the temperate zones of the earth.

In winter, man secures comfort by wearing heavier clothing, by supplying buildings with heat and conserving that heat by keeping windows and doors closed. In summer artificial heat is cut off, doors and windows are opened, and lighter clothing is donned. Practically, human comfort depends more on maintaining a proper temperature balance between body and surrounding air than upon any other factor in the atmosphere. The ventilation requirement of greatest importance, both winter and summer,

is, thus, a control of temperature within a short range of the optimum. This is a twentieth century conception of ventilation needs. As we shall show, the conception of ventilation was quite different in 1870. As our understanding of the subject has changed, so have the methods used to secure ventilation.

THE CARBON DIOXIDE BOGEY

It is difficult to trace chronologically the theories respecting the underlying badness in used or confined air. The pendulum has swung back and forth. Before the discoveries of oxygen and carbon dioxide as components of air the mysterious poison theory was uppermost. When the physiology of respiration—the conversion of a portion of oxygen into carbon dioxide in the lungs—came to be understood, leading scientific thought seized upon carbon dioxide as the culprit. Knowing that oxygen was essential to life and that an animal would soon die from suffocation in an air-tight box, it was assumed that the diminution of oxygen and the increase in carbon dioxide in the ordinary room was also harmful. This was the situation in the 70's, and as it is our purpose to trace the ventilation problem during the lifetime of the *American Public Health Association,* it is not to the point to go back of this date.

The whole thought of ventilation in the 70's was to prevent people from being poisoned by their own exhalations.

A lay writer, John Brown, expressed the popular sentiment of the time in the following sentence, "Everybody knows that carbonic acid gas is fatal, but everybody does not realize how much we suffer from the moderate doses of it which we are constantly inhaling."

This sentiment prevailed to a widespread degree though past the 80's and 90's and in fact reveals itself in the expressions of some even at the present day. It is strange how persistently belief in the harmful nature of carbon dioxide has remained in lay and scientific thought. It is like the memory of a long lost friend that will not down. The strangeness of this persistence is referred to because as far back as 1842, Leblanc showed that an animal can breathe an atmosphere containing 30 per cent of carbon dioxide for three-quarters of an hour, so long as the percentage of oxygen was kept around 70, and then recover on removal to the normal atmosphere. Furthermore, he showed that the increase in carbon dioxide which takes place in theatres and crowded places is surprisingly slight relatively, and that the reduction of oxygen is quite insignificant.

In 1857, Claude Bernard performed a variety of experiments with animals, all tending to show that the increase of carbon dioxide in commonly occupied places was far below a lethal or even an irritating limit. In 1863 Von Pettenkofer proclaimed the harmlessness of carbon dioxide as it occurred in the ordinary occupied room. He did not believe that rebreathed air was a poison in the ordinary sense of the term, but he did believe that vitiated air increased susceptibility to disease. Von Pettenkofer unwittingly fastened odium upon carbon dioxide by pro-

posing the carbon dioxide test as an indication of the amount of other impurities in the atmosphere.

It is probably because the CO_2 test has become so universally used in ventilation studies that a belief in the harmfulness of the gas itself still persists.

ORGANIC EFFLUVIAL POISON

Carbon dioxide slowly but surely lost its adherents during the 80's. Perhaps experience had proved that its harmfulness in small quantities was unfounded. Crowded rooms still caused discomfort, and some reason for the annoyance had to be advanced. Nothing was known about "organic effluvia," and it was consequently elevated to the throne of iniquity. It was splendidly fitted for the part. It possessed an element of mystery that was tremendously satisfying. Greenleaf, a civil engineer, wrote in 1885 as follows:

"Now the best opinion is in favor of the view that the vitiating of air by human beings, so far as it is made unfit for use, is due chiefly to the organic matter given off."

Griscom had hinted at this in 1850, " . . . On the whole it would appear that sulphuret of ammonia is the morbific agent exciting typhus fever, sulphuretted hydrogen being the pestilential virus producing yellow fever, and the bilious remittents, and agues of tropical climates." Griscom refers to no less an authority than Liebig as having found ammonia in the atmosphere, and he "has well nigh established the identity of malaria with sulphuretted hydrogen."

The organic poison theory of bad air found a ready champion in Brown-Sequard and d'Arsonval, who reported in 1887 that expired air from healthy men and dogs had the power of producing toxic phenomena. They injected the washings from the trachea and condensed moisture from the breath into animals, and death ensued in from a few days to several weeks. They were convinced of the presence in the expired breath of a volatile poison in the nature of an organic alkaloid.

They performed also another experiment which has had great weight in inducing people to believe in the dangers of vitiated air. They confined rabbits in a series of tight metallic boxes connected by rubber tubing through which a constant supply of air was drawn. The animal in the fourth cage breathed the air passing through the other three cages. The third animal breathed the air of the first two, the second animal the air from the first, and the first animal received air directly from outside. In a few hours the animal in the fourth cage died. Some time later the animal in the third cage died. The first two animals usually remained alive. Inasmuch as the air in the last cage rarely showed by analysis more than from 3 to 6 per cent of the CO_2, they concluded that death was caused by a volatile poison. This belief was supported by the fact that when the air from the third cage was passed through sulphuric acid before entering the fourth cage, the fourth animal remained alive, whereas the third animal died. The acid, they said, absorbed the poison and thus protected the fourth animal.

Brown-Sequard and his colleagues succeeded in stimulating a great amount of experimentation, the results of which largely failed to confirm their findings. For instance, Haldane and Smith in 1892 confined a person in a small chamber until the vitiation was from ten to twenty times as great as in the most crowded and worst ventilated building, and yet there was no perceptible odor or sense of oppression.

Billings, Mitchell and Bergey repeated Brown-Sequard's work and were unable to secure any evidence which pointed to the presence of a volatile poison in the expired breath. For instance, six rabbits were kept in a series of bell jars for 42 days, the proportion of CO_2 in the last two jars much of the time being from 4 to 7 per cent and of oxygen from 12 to 16 per cent. *None of the animals died or were seriously ill. "Those in the first three and in the fifth jar gained in weight, those in the fourth and sixth lost slightly in weight."* They concluded that death occurred in the latter jars of the series when the CO_2 was increased to 12 or 14 per cent and oxygen was diminished to 4 or 6 per cent. Death did not occur when the CO_2 remained less than 10 per cent unless the oxygen was reduced to below 7 per cent of the mixture. They believed the sulphuric acid did nothing more than to interpose a resistance in the circuit so that leakage of air took place around the connections to the last jar, thus reducing the CO_2 content and increasing the proportion of oxygen.

The idea that an organic poison actually exists in the exhaled breath was not by any means dispelled in 1895. In fact, as late as 1911 Rosenau and Amos believed that they could demonstrate this fact by producing anaphalaxis in animals following the injection of small quantities of condensed matter from the breath of other animals, to whose breath the animals under experiment had previously been exposed for some time. Subsequent work along this line independently by Leonard Hill, Weismann and Lucas have failed to substantiate Rosenau's claims.

TEMPERATURE AND MOISTURE GAIN A HEARING IN 1895

While there is occasional reference prior to 1880 to high temperature and moisture as being undesirable and discomforting, the references are given half heartedly. Heat and moisture were merely looked upon as accentuating the poisonous effects of organic matter and carbon dioxide. They were not given primary weight of themselves. In 1883 Hermans was bold enough to suggest that chemical impurities were insignificant as causes of discomfort in occupied rooms and that bad ventilation was in reality due to excessive temperature and moisture. Like so many pioneers, Hermans' voice carried little weight at the time. It was not until 1895 that the dawn of a new conception of ventilation appeared. Evidence of this is found in the refreshingly clear statements of Billings, Mitchell and Bergey. After a review of the experimental work up to that time and a repetition of much of this work under their direction they came forward with the statement that it is *"very improbable that the minute quantity of organic matter contained in the air expired from human lungs has any deleterious influence upon men who inhale it in*

ordinary rooms, and, hence, it is probably unnecessary to take this factor into account in providing for the ventilation of such rooms."

We cannot forbear quoting further from these authors, for their pronouncements in 1895 are so closely in agreement with our understanding of the situation today after 25 years of further experimentation.

"The discomfort produced by crowded, ill-ventilated rooms in persons not accustomed to them is not due to the excess of carbonic acid, nor to bacteria, nor, in most cases, to dusts of any kind. The two great causes of such discomfort, though not the only ones, are excessive temperatures and unpleasant odors. Such rooms as those referred to are generally overheated, the bodies of the occupants, and, at night, the usual means of illumination, contributing to this result."

They conclude from their researches that some of the theories of ventilation prior to this time were either without foundation or were doubtful and that the problem of securing comfort and health in inhabited rooms was one of preventing or disposing of various dusts, of regulating temperatures and moisture, and of preventing the entrance of poisonous gases such as carbon monoxide "rather than upon simply diluting the air to a certain standard of proportion of carbonic acid present."

We have dwelt at length on the theories of ventilation because of their importance, and we shall return to this question later on. The reader must appreciate the fact that so long as people were worried to death over a mysterious poison in the air of a crowded room and over a distressingly slight excess in carbon dioxide over that in the outside air, they could not think of ventilation except in terms of flushing rooms with air so as to keep down the concentration of these supposedly evil substances. They saw poison and suffocating gas and their remedy for these supposed conditions was the proper one. They failed to secure universal comfort because in spite of ample air dilution they made no special effort to keep rooms cool.

THE SERVICE PERFORMED BY STEAM HEAT AND THE MECHANICAL POWER

Because the word *fail* was used in the previous paragraph, it must not be taken for granted that progress in ventilation was lacking from the 70's to the 90's. On the contrary, there were well-defined advances during this period. In the 50's and 60's steam radiators, both direct and indirect, made their appearance, and these devices taken in conjunction with the blower fans made possible voluminous flushing and aeration with tempered air which had not been satisfactorily worked out prior to this time.

To be sure, the House of Commons in England was ventilated by a crude centrifugal fan back in the 18th century and later in 1846 the U. S. Custom House in Boston installed a steam-driven fan. But this was in the perplexing days of trial. Prior to 1870 fans were confined largely to industrial plants to remove dusts and gases. It was after 1870 that mechanical ventilation began to find its way into school buildings and larger places of assembly. It performed a distinct service, because air flushing was an improvement over conditions that had existed, and

temperature could be controlled within narrower limits. School buildings, for instance, formerly depended on natural draft or warm air furnaces. Ducts were provided for supply and exhaust, but they were small and poorly located. In New York City today one can find old schools with exhaust registers 1 or 2 ft. square placed in an out-of-the-way corner, the exhaust ducts frequently being in or near the cold exposed wall with a mighty slim chance of air leaving the room through such a portal. The responsibility for this error has been laid at the door of the architect, it being charged that he was more interested in the beauty of arrangement than in the utility of the ventilating equipment. Perhaps the architect was at fault, although he may have an alibi. Perhaps the school commissioners insisted on putting those unsightly registers in an inconspicuous place. They succeeded at least. Regardless of who was at fault, ventilation did not have a ghost of a show with such treatment.

The power fan did force some air through the rooms, and a badly located register could not altogether prevent this. At times this air flow cooled the densely populated rooms, but as a rule room temperatures still fluctuated widely and overheating was all too common.

Thus, while positive air flow helped to keep down carbon dioxide and to lower temperatures, it did not insure cool rooms at all times, the cooling being purely incidental. Nor did air flushing keep down odor.. As Billings pointed out in 1893, the smell of organic matter may not be perceptible when the CO_2 content due to respiration amounts to an excess of 5 parts per 10,000; and again, odor may be very decided when the CO_2 does not exceed 3 parts. It depends on temperature and moisture and the amount of diffusion going on, and odors do not diffuse as rapidly as CO_2.

Air flushing alone will not always remove odors. As Pettenkofer remarked, "If there is a dung heap in a room, it must be removed. It is no good trying to blow away the smell." It is much more effective to wash dirty bodies and dirty clothes with water than with air. In warm rooms people perspire and vaporize the odorous salts and substance in body and clothes. In cool rooms odors are less noticeable.

STANDARDS OF AIR FLOW

Again do we feel the urge of apologizing for our rather free criticism of the shortcomings of ventilation in the period of 1870 to 1900. As has been said, progress was measurable during this period, and against the background of the 50 preceding years it was a very marked advance. The air flow standard of 30 cu. ft. per min. per person was well established at this time. Back in 1824 Tredgold, an English engineer, had the temerity to propose that a certain amount of air change was necessary and he very modestly suggested a standard of 4 cu. ft. per min. per person. In 1835 Dr. Reid, who was identified with the ventilation of the House of Commons, looked upon 10 cu. ft. of air as essential. In 1857 the Barrack Commissioners of England fixed upon 20 cu. ft. as the proper amount. Morin (1873) regarded 8 cu. ft. as necessary in schools

for children under 15, and 14 cu. ft. for children over 15. For theatres he recommended 25 cu. ft. and the same for army barracks at night.

It remained for DeChaumont in 1875 to interpret good ventilation as demanding less than two parts of CO_2 per 10,000 above the normal. To maintain this level of 6 parts CO_2 requires 50 cu. ft. per min. per person. This was the standard adopted by Dr. Parkes and many English writers. Billings looked upon 30 cu. ft. as a minimum, but with provisions for increasing this to 60 should the occasion demand.

The standard of 30 cu. ft. per min. per person for schools met general acceptance in this country and is incorporated in the statutes of a number of states today.

The period from 1870 to 1890 thus may be credited with giving us our present day standards of air flow, which are widely used by engineers. While superior to preceding standards they have led us astray by focusing so much attention on air volume as to obscure the important matter of temperature, and the desirable feature of fluctuation in air flow.

TWENTIETH CENTURY CONTRIBUTION TO VENTILATION

The last 20 years have beeen characterized mainly by an increasing respect shown to the thermal factors of the atmosphere with a corresponding diminishing importance attached to the chemical constituents.

That the badness of bad air is due more frequently to over-heating and excessive moisture than to any other cause was brought out in most convincing fashion in 1905 by Flugge and his pupils, Heymann, Paul and Ercklentz. They placed normal people in a cabinet of 3 cu. m. capacity. In four hours' time the CO_2 had risen to 1.0 and 1.5 per cent (100 and 150 parts per 10,000 as compared with 4 parts in outdoor air.) No illness or discomfort was felt so long as the temperature and humidity were kept down. When the temperature was allowed to rise, discomfort was felt. At a temperature of 75 deg. and 89 per cent relative humidity and a CO_2 content of 1.2 per cent the subjects felt very uncomfortable. Breathing fresh air from outside gave little, if any, relief. A person on the outside experienced no discomfort on breathing the air of the chamber through a tube. A person entering the chamber experienced immediate discomfort. When a fan was set in motion within the chamber, stirring up the air, there was immediate relief even with no fresh air from outside. The layer of warm, moist air next to the skin was set free and distributed more thoroughly over the entire chamber.

The interpretation of these experiments is that it is not the chemical constituents of used air which cause unpleasantness to those in the room. It is wholly the increasing heat and moisture which are responsible for discomfort.

This work has been repeated a number of times with the same results. Leonard Hill and his English colleagues, Flack, McIntosh, Rowlands and Walker, in 1913, published their account of a repetition of Paul's work with complete confirmation. The New York State Commission on Ventilation found substantially the same results in 1914 and 1915,

and its work was on a larger scale and covered longer periods of time. Instead of hours, subjects were kept under varying atmospheric conditions for the entire day, day after day and for as much as six successive weeks. The Commission found that overheating, even slight overheating, 75 deg. fahr., increases discomfort and brings on premature fatigue. It increases body temperature and the heart rate. When working on the piece work basis, a bonus being paid for quantity of work done in addition to a flat wage, subjects did 13 per cent less work at 75 deg. than at 68 deg. *It mattered not whether the chamber was bathed in fresh outside air (45 cu. ft. per person per min.) or whether it was sealed so tightly that the CO_2 rose to 30, 50 and even 130 parts per 10,000 so long as the room temperature was kept down to 68 deg., the subjects did just as much work and were quite contented.* But let the temperature rise to 75 deg., even though the humidity did not exceed 50 per cent, work slackened, perspiration began and discomfort arose. The only indication of any unfavorable effect of stale air—that is air undiluted with fresh air—was in the slightly lessened food consumption, and this was most noticeable in the first or second week, largely, if not entirely, disappearing in the following weeks. Physical efficiency was but slightly affected.

The above are the contributions of the laboratory man, physician, physiologist, chemist and physicist. The practical ventilating engineer, architect and layman may look askance at those experiments on a few people in small rooms, suspecting that the same laws will not hold true with many people in large rooms. This question has often been propounded. Even if these short-time experiments prove that air vitiated by the products of human exhalations is not harmful, is it not possible that a long-continued stay under these conditions, even though the air is cool, will eventually affect health? The answer to this is that the best of experimental evidence shows that the chemical constituents of vitiated air of ordinary occupied rooms are innocuous to health, and until evidence is produced to the contrary it is entirely logical and scientific and a matter of common sense to abide by these findings.

HOW THE NEWER EXPERIMENTAL EVIDENCE HAS AFFECTED VENTILATION PRACTICE

Ventilation has inevitably been closely associated with heating. In the days before the mechanical blower and the steam radiator, buildings were heated by stoves and furnaces. The stove heated the single room. The furnace was an elaboration of the stove with the addition of a jacket through which air passed and then ascended through ducts to a number of rooms. The temperature of the air had to be high, that is above 100 deg. fahr., to do its work. It was found that the distribution of this heated air was seriously interfered with by outside winds. Furthermore, the great heat of the air caused an annoying sensation of dryness to occupants.

The steam and hot water radiator provided a convenient means of direct heating. This solved the heating question for the time being, but

left the matter of ventilation to be dealt with independently. Air ducts were then added, and in the exhaust ducts it was customary to place steam pipes or burning gas jets to accentuate upward air currents. These ducts were usually too small in area to be effective, and consumed a great deal of fuel.

The next step was the introduction of the steam-driven fan which forced air over heated coils or radiators. This method was used to ventilate and it was also depended upon to do the entire heating. This again introduced the same objections associated with the hot air furnace.

A modification of this plan was the placing of direct radiation within the room to supply heat and the use of blowers and indirect heating coils to furnish tempered air for ventilation. This is the arrangement most common today.

The introduction to rooms of large volumes of air in order to maintain the standard of 30 cu. ft. per min. per person necessitated the use of more spacious ducts, for otherwise the velocity of the entering air was so great as to cause annoying draughts. Larger exhaust ducts naturally followed.

Improvement took place in air distribution, which involved more careful placement of inlet and exhaust registers. Engineering details likewise advanced as the use of this equipment became more general. The steam-driven fan gave way in many instances to the electrically-driven fan, and direct connected motors displaced the belt connections between motor and fan.

Automatic temperature control replaced manual control, and today the thermostat is an essential part of indirect heating and is quite common in connection with direct heating.

We can see today many efforts to effect improvements in air distribution. The exhaust fan was added to assist the plenum fan in keeping the flow of air confined to the channels provided and also to assist in the proportioning of this flow over the building. Efforts have been made to maintain air circulation wholly by exhaust fans, the plenum fans being omitted and air being admitted to each room through apertures in the outer wall, the air temperature being raised by contact with the direct radiation. The passage of cold air over the radiators resulted frequently in frozen pipes, particularly when steam pressure was low. Furthermore, a strong suction from the exhaust duct at times drew air from open doorways or other apertures and caused short circuiting with consequent reduction in the aeration of the room as a whole.

INDIVIDUAL ROOM DUCTS FOR TEMPERATURE ADJUSTMENT

When the sun is shining the rooms in its path are warmed more than rooms on the shaded sides of a building. Consider the effect of this when the temperature of the indirect air supply is the same for all rooms. Some rooms become overheated, and to correct this condition the temperature of the air supply to the sun-exposed rooms should be reduced. The practice most commonly used now is to have a separate cold air supply

controlled by dampers operated by thermostats in each room. A room that is too warm will have its warm air supply cut down by the damper and some cold air will be admitted to lower the temperature. This involves individual ducts to each room from the hot and cold air chamber in place of the single trunk duct leading up to a certain section of a building with branching ducts to each room.

HUMIDIFICATION

The use of humidified air was ushered in just prior to the 20th Century and was prompted by both hygienic and economic considerations. There are many industrial processes, such as weaving, which require a relatively high moisture content in the rooms. Then also mechanical ventilation with its large volumes of air produced so much evaporation from the bodies of occupants as to cause annoying sensations of dryness. When cold outdoor air is heated to 60 or 70 deg., the relative humidity changes from the 70's and 80's to 20 and 30 per cent. The heating of air reduces its relative humidity. When air is saturated with moisture (100 per cent relative humidity), a cubic foot contains at 30 deg. fahr., 1.94 grains of water, at 70 deg. fahr., 7.98 grains Thus when outdoor air at 30 deg. fahr. and 80 per cent relative humidity is taken into a building and heated to 70 deg. fahr., the resulting relative humidity is only 19 per cent. The actual quantity of water (absolute humidity) present is the same but at 30 deg. this amount represents 80 per cent of all the moisture that the air could contain, whereas at 70 deg., this same amount represents only 19 per cent of what the air could contain. Warm air at a low relative humidity tends to extract moisture from its surroundings. It was to correct this condition that humidifying pans and steam jets were added to ventilating equipment. These have been superseded largely by air washers which not only add moisture to the air but also remove dust. Many installations today include air washers as an integral part of the ventilation equipment. This involves also some means of automatic humidity control.

CHANGES IN HEATING PRACTICE

In 1870 the stove and hot air furnace were most widely used for heating and ventilating. Hot water heating was being advocated for houses and steam was confined mostly to factories. Gravity ventilation in schools with supply ducts and heat acceleration in the exhaust ducts was not uncommon, but most schools had no provision whatever for ventilation. Mechanical ventilation was unknown in school buildings.

Thirty years later conditions had changed. We have an interesting picture of school ventilation in 1899 from the report of John Gormly. The Board of Education of the city of Philadelphia then had supervision over 447 schools and found:

13 were heated by direct steam radiation.
38 were heated by direct and indirect steam radiation.

152 were heated by hot air furnaces.
14 were heated and ventilated by blowers and steam coils.
4 were heated and ventilated by batteries of hot air furnaces, through which blowers forced air into the rooms, the blowers being operated by gas engines.
None were heated by hot water.
Many of the remaining were heated by stoves or by combinations of stoves and furnaces.

In schools heated by hot air furnaces and blowers driven by gas engines there was much complaint of the smell of hot oils, gas and noisy machinery. There was much vibration in the rooms above the machinery. Evaporating pans were provided for humidification, but were rarely used.

In 1921, the stove heated school building has disappeared, at least in the larger cities. Furnace heat in large schools is a rarity. Gravity ventilation with direct heat and the various mechanical systems—plenum, combined plenum and exhaust, and exhaust alone with direct-indirect heat—are in most common use.

RECIRCULATION

As may be appreciated from the foregoing, the development of ventilation has proceeded from the simple to the complex. Each year has witnessed new and more elaborate equipment, and along with this advance has gone mounting cost. To offset the increasing expense there has been proposed a recirculation of the air within the building. Obviously much fuel will be saved if, instead of cold air from the outside, the warm air drawn from the building can be sent back through the supply ducts. Presumably this proposal would have been hooted with scorn in the days when carbon dioxide and organic effluvia reigned in their poisonous glory. However, Walter Snow urged recirculation of factory air as far back as 1891. Where the cubic space is great, this is an entirely feasible plan, for natural leakage through walls and around doors and windows is sufficient to prevent any unusual accumulation of odorous material in the air. Recirculation has worked out with apparent success and appreciable saving in fuel in the gymnasium of the Y. M. C. A. Training School in Springfield, Mass. This method has much to commend it in places where the air space is ample and the vitiating factors—from human beings, illuminating gases and industrial processes—are relatively slight.

The best example which has come to our attention concerning this practice in a school building was described by Prof. Larson in 1916. The Wisconsin High School at Madison was ventilated with apparent success by utilizing only the air which was returned from the rooms, and none from outside. The air supply was cut down to about 15 cu. ft. per person per min. Room temperature ran from 65 to 67 deg. and relative humidity around 60 per cent. All air was passed through a washer before readmission to the rooms. The velocity of air entering the rooms was around 600 ft. per min., or double that in usual practice. Carbon dioxide ran as

high as 20 parts per 10,000 and averaged above 10. By this method there was a saving in fuel of from 40 to 50 per cent. Larson makes this commentary, "Ventilation by recirculation is both efficient and economical. At the end of a year's run the teachers are almost unanimous in their praise of the system."

The point to be borne in mind in connection with this work is that this building was occupied only to about 75 per cent of its capacity, and the cubic feet of space per student was more than 350 as compared with a standard of 200 cu. ft. in grade schools.

Thus with buildings having a large ratio of air space per occupant and with occupants moving about at frequent intervals during the day, recirculation offers an attractive method of ventilation. For crowded rooms or buildings it has disadvantages. Thus the New York State Commission on Ventilation found that recirculating the air of a single class room was associated with physical discomfort and odor. Some outside air had to be admitted to reduce odor. Air washing intensified odor, and recirculation was more successful with the air washer shut down. Ozone was used to reduce odor, but this likewise aggravated rather than helped the situation. The combination of ozone with the room odor was worse than the room odor by itself. Odor was less noticeable with humidities around 40 per cent or below than with higher humidities.

THE NEWER VENTILATION STANDARDS

The primary purpose of ventilation in 1870 was to remove carbon dioxide and organic effluvia. This view had not altered greatly in 1881, for Leicester Allen, a mechanical engineer, wrote as follows:

"The end and purpose of ventilation is, therefore, to remove from inclosed spaces these emanations (referring to organic matter given off from human bodies and also carbon dioxide) together with any other poisons which may have become added to them either from leaks in gas pipes or fixtures, from soil pipes, from defects in water closets or from any other source."

The carbon dioxide content was the measure of ventilation. Allen states that "When by the presence of people in an inclosed space the percentage rises from 6 to 7 parts (of CO_2) in 10,000, the ventilation is bad. When the proportion is less than this the ventilation is good."

The change in our conceptions of ventilation since the 19th century may be judged from Winslow's definition of good ventilation as recorded in Park's Public Health and Hygiene published in 1920.

1. "The air should be cool, but not too cold."
2. "The air should be in gentle, but not excessive motion, and its temperature should fluctuate slightly from moment to moment."
3. "The air should be free from offensive body odors."
4. "The air should be free from poisonous and offensive fumes and large amounts of dust."

Notice the order of these requirements; observe the omission of the CO_2 standard and the air supply standard of 30 cu. ft. per min. per person and take note of the avoidance of set values in the definitions. The physical and cutaneous factors—temperatures and air movement—take precedence over the chemical and respiratory elements. There is a suggestion of variability in temperature and air motion as a desirable condition as opposed to uniformity in these conditions.

From the standpoint of the engineer who is called upon to design ventilating equipment, this definition will be criticized as too general.

The engineer needs something more specific. What temperature? How much air supply? What limits of fluctuation in temperature and air movement?

Winslow suggests a temperature between 65 and 68 deg. in the office and school room and even lower in the factory and open air school where physical work is performed or heavier clothing is worn.

No mention is made of the degrees of fluctuating air movement. Further experience must determine this figure in specific terms. The engineer may, however, act upon the theory that variability is wanted rather than uniformity, and in planning his fan equipment this point must be borne in mind.

As to the air volume and rate of change, there is no dogmatic assertion. This is a matter that must be worked out for specific purposes. In commenting on this point, Winslow states: "The fact is that while 30 cu. ft. per min. may be set as a rough standard often desirable to attain, it may be insufficient in certain cases and in many other cases may be an unnecessarily high figure. Hygienists must insist on standards of air conditions actually maintained rather than on any arbitrary mechanical standard assumed to be capable of maintaining them. Any system which does not guard against overheating is inadequate, however large a volume of air may be forced into a room."

As to humidity Winslow agrees that high temperatures combined with high humidities are particularly harmful. He does not lay stress on the hygienic influence of humidity at lower temperatures.

DEFICIENCIES IN EXISTING LAWS AND ENACTMENTS ON VENTILATION

How do existing statutes and regulations conform to this definition?

They differ primarily in the air supply standard. The New York State Law passed in 1904 recites that the "Commissioner of Education shall not approve any plans for the erection of any school building in third-class cities or incorporated villages or school districts, the cost of which building or addition exceeds $500, unless provision is made therein for assuring at least 30 cu. ft. of pure air every minute per pupil, and the facilities for exhausting the foul or vitiated air therein shall be positive and independent of atmospheric changes."

The Pennsylvania law enacted in 1905 requires that school-houses "shall provide for an approved system of indirect heating and ventilation

by means of which each classroom shall be supplied with fresh air at the rate of not less than 30 cu. ft. per min. for each pupil and warmed to maintain an average temperature of 70 deg. fahr. during the coldest weather."

In 1914 the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS promulgated a set of minimum ventilation requirements for public and semi-public buildings, which were recommended as model compulsory heating and ventilating laws. Those which applied to schools specified that:

> A positive supply of outdoor air shall, while school is in session, be provided the following rooms, and the quantity of this positive supply shall be equal to, or in excess of the following minimum requirement per occupant per hour. The standard for class rooms, study and recitation rooms was given as 1800 cu. ft., or 30 cu. ft. per min. per person. Gravity indirect heating and ventilating systems are accepted for small schools not exceeding 8 class rooms in localities where it is impossible to secure proper motive power, provided such gravity systems are capable of supplying at least 30 cu. ft. of air per min. for each pupil.

San Francisco has a section in the building law which requires "30 cu. ft. of pure air per min. for each occupant," of all buildings designed to be used in whole or part, as public buildings, public or private institutions, schoolhouses, churches, public places of assemblage or places of public resort, including theatres and audience halls, factories workshops and mercantile establishments.

The Sanitary Code of Chicago adopted in 1911 is very comprehensive and specific in the matter of ventilation. Temperatures exceeding 68 or 70 deg. are prohibited in certain types of occupied buildings during the heating season. In street cars, elevated and railway trains a range from 50 to 75 deg. is permitted. There is nothing said about the fluctuation in air flow. There is provision requiring the removal so far as practicable of all noxious or poisonous fumes or gases or dusts arising in connection with a manufacturing process.

The Chicago Code specifies floor and cubic space requirements for different types of buildings as well as quantities of air to be passed through them. Air flow requirements vary from about 1000 to 2000 cu. ft. per hour per person. A maximum CO_2 standard of from 10 to 12 parts per 10,000 is specified.

There is also included a section requiring that the relative humidity of certain enclosures shall not be less than 40 or 45 per cent and not more than 80 to 85 per cent. The lower limit of humidity would seem to be unjustified from existing experimental evidence.

The intent of the above laws and regulations is to provide good ventilation. A criticism that may be levelled against them is that they are too specific in the matter of air flow. There is not sufficient justification for demanding a continuous air flow in the proportion of 30 cubic feet per minute per person for so varied a group of buildings.

For instance, it is entirely feasible to maintain good ventilation in such places as school rooms, when equipped with:

a. Direct radiation,
b. Window deflectors,
c. Spacious gravity exhaust ducts on the inside wall,
d. Readily adjustable windows,
e. Readily adjustable temperature control, automatic or hand.

This type of ventilation shows from the experience of the New York State Commission on Ventilation great promise in providing cool atmospheres, with fluctuating air movement, free from offensive odors and large amounts of dust. And yet under the above rules and regulations this method is completely ruled out because of the positive air flow requirements of 30 cubic feet per minute, which can be produced only by mechanical means. Recirculation likewise is rendered impossible by many of these laws and regulations.

Notwithstanding the inconsistencies as previously pointed out, there is a uniformity of opinion on certain fundamental ventilation requirements which may be grouped as follows:

1. Except in rooms occupied by a few persons only, ventilation solely by means of windows located on one wall of a building is unsatisfactory. Consequently a schoolroom with window exposure on one side only and without exhaust ducts cannot be ventilated in a satisfactory manner.

2. Assembly rooms, audience halls, theatres, court rooms and other places where people congregate in such large numbers that the floor space per person is not more than 10 sq. ft. cannot be satisfactorily ventilated during the heating season except by means of mechanically propelled air.

3. During the summer time moving picture theatres and other places where people congregate to a similar extent demand mechanically propelled air for comfort. This may be accomplished by numerous circulating fans distributed at various points or by blower fans sending cool air through the enclosure or by a combination of the two. The cooling of the air is done in various ways—by means of an air washer, ice or brine coils.

As to the ventilation of schoolrooms there exists a divided opinion. As already stated, the work of the New York Commission indicates certain undeveloped and hitherto unrecognized advantages of window ventilation with gravity exhaust. There is no question that the statutes regarding air flow for this type of room should be altered so as to permit the use of window ventilation.

Perhaps the most unsettled question at the present time is that of humidity. In rooms which are not overheated is humidification necessary for health and comfort? There is much to be said pro and con. The fact remains that we have much still to learn on this point. High humidity with high temperature is detrimental to health. Whether low humidity with temperatures below 70 deg. is objectionable is a matter still awaiting convincing evidence.

The study of sanitary insignificance of bacteria has been a new development since 1870. While within the period of 1880 to 1910 there was considerable perturbation over the danger of bacteria floating about in the air, as evidenced by our past efforts at gaseous fumigation, their significance from a ventilation standpoint is almost nil, a possible exception being in hospitals and operating rooms and certain industrial processes such as the handling of hides and furs which may contain anthrax spores. There are other ways far more effective than ventilation for limiting the spread of infectious disease.

Dust is a factor to be reckoned with in industrial establishments and in mines. It may be annoying in other buildings, but its significance below the limits of sensibility is slight. The suppression of dust in school, theatre and mercantile establishments is largely a matter of floor cleanliness or location of air intake, rather than of air flushing.

MEETING THE PRESENT DAY VENTILATION NEEDS

Aside from the special ventilation problems of industry involving dust and fume removal, the greatest need today in homes, offices, schools, barracks, prisons and hospitals is for facilities for maintaining a proper temperature range. This means easily adjustable windows, window deflectors, direct radiation, ready control of heating unit either automatically or by hand, and where it is impossible to secure cross ventilation by windows there should be provision for exhaust ducts with dampers near the interior of the room. It must be the aim of the engineer and the architect to provide people with the means to keep comfortably cool. We have had enough of the overheated radiator which cannot be turned off, the traversing of an occupied room by uninsulated steam pipes whose heat it is impracticable to cut off, the window which will not budge. More attention must be paid to these details in order to improve indoor air comfort in the less crowded inclosures.

In schools which are mechanically ventilated we must discard the 30 cu. ft. standard and also give up the idea that it is necessary to send air to every point in the room. The extremists in 1870 wished to flush a room as a body of policemen would clear a hall, pushing all before them. This was ventilation by displacement. We now ventilate by dilution. It would seem that still better results could be accomplished in schools and auditoria by substituting surface skimming in place of flushing. We must also substitute the intermittent or ejector type of air flow for the monotonous uniformity which now characterizes mechanical ventilation. This will mean changing the inlet register from its accustomed place on the inner wall to the space between the direct radiation and the outer wall. The course of air flow will be upward across the upper levels of the rooms and down the inner wall to the outlet. Windows may then be opened with impunity, the mechanical air flow creating an aspirating effect which will draw in the cold outside air and send it over the room. By this means we shall harmonize the conflicting interests of the teacher who wants the window open and the janitor who wants it kept closed.

The occupants of the room, it is true, will not be uniformly perflated. At times they will sit in an atmosphere higher in CO_2 than under the old arrangement, but they will not sit in direct draughts of unvarying air currents, and consequently they can stand lower temperature. It is almost impossible, in many fan ventilated schoolrooms to secure comfort at 68 deg. temperature, because of the appreciable air flow through the room, which cools by convection and evaporation. Seventy and even 72 deg., are more comfortable. Direct the circulating air over the upper portions of the room and temperatures of 68 and even 65 deg. are not uncomfortable. Dryness will not be noticed with overhead air circulation. Air from the windows will aid in giving a pulsating or fluctuating motion to the mechanically propelled air stream, which will be both pleasant and stimulating.

The adoption of this general plan for the ventilation of a theatre has been recently described by Samuel R. Lewis. Its adaptation to auditoria and rooms similar to the ordinary schoolroom holds out great promise.

Another weakness of present day ventilation is the great and unnecessary waste of fuel. There is no reason other than the 30 cu. ft. of pure air standard which militates against the use of recirculated air for many types of enclosures with ample space provisions, such as, gymnasia, churches, large assembly halls sparsely occupied, stores and factories. The same air might just as well be recirculated with a consequent saving in fuel. The natural dilution that takes place will meet all hygienic requirements so long as the temperature is kept down below 70 deg., and if this is done humidification will in most cases be unnecessary. Odors are less pronounced in a dry atmosphere than in one of high relative humidity. In short, let us be free to apply the remedy that will insure maximum comfort, health and efficiency to each specific case without being prevented from so doing by well intentioned but overzealous requirements of air flow.

The deficiency of ventilation in warm weather is being overcome by the increasing use of the disc fan for evaporative cooling and the use of the air washer in theatres for lowering temperature by vaporization. This is a distinct advance of recent date in ventilation practice.

One further unsolved problem of the present day is the ventilation of railway coaches, parlor cars and sleepers during winter weather. This is almost wholly a matter of temperature control. There has been a signal failure in providing passengers with air comfort while travelling, and the journey without most uncomfortable overheating day and night or else extreme underheating is a rare one.

THE CONTRIBUTIONS OF VENTILATION TO BETTER HEALTH

Have advancements in ventilation contributed anything to human happiness, health and longevity in the last 50 years?

Most decidedly, yes!

The unnecessarily hot, damp factory which fatigued operatives prematurely and reduced resistance to colds and pneumonia has largely disappeared in this time. Some form of ventilation is now looked upon

as an integral portion of many types of buildings. It was purely an incidental factor 50 years ago.

The dust and fume exhaust system has without question contributed to our declining tuberculosis rate.

Today the open window by day, and more particularly at night, is known and utilized by many more than formerly. The lack of ventilation in living quarters is responsible today for much of the tuberculosis in the land of our good friend and ally, the French Republic. The American doughboy who sojourned overseas can tell you that good ventilation is an adjunct to health.

Air comfort prevails in theatres and assembly halls and in schools in an increasing measure each year, and pneumonia is losing its victims with each advance. Good ventilation as defined in the terms of 1921 is a big factor in the war on respiratory disease. As Dr. S. Josephine Baker has shown, cool rooms with fluctuating temperature and air flow specifically proved their value in reducing colds and respiratory affections among New York City school children in 1916 and 1917.

The mechanical blower and the steam radiator have been a godsend to the factory operative and the girl at the department store counter. Even though located a long distance from the open window, these workers in our immense 20th century buildings may be supplied with cool revivifying air. There is no longer any excuse for adding high temperature fatigue to the normal fatigue of physical exertion.

To permit ventilation to aid still further in the cause of human welfare, there remains only the reframing of our laws, rules and regulations in the light of recent experience, so that ventilation essentials will be insisted upon and non-essentials be relegated to their proper subordinate position.

Just so soon as the hygienist incorporates in clear and understandable phraseology the health, comfort and efficiency demands of ventilation, attributing to each element its proper weight and no more, and so soon as the weight of an enlightened public opinion transforms these requirements into legal enactments, we can depend upon those who build our enclosures—the engineer and the architect—to provide for us the condition desired.

DISCUSSION

F. R. STILL: I think the author is not acquainted with the most modern investigation as developed by the research work in our own Laboratory. He is apparently unaware that it is well established what the wet bulb should be and that it should vary according to the variations of air movement.

DR. E. V. HILL: I think it is for us to continue to forge ahead and work out some definite recommendations regarding recirculation of air, infiltration and similar problems rather than trying to answer statements regarding open windows.

R. A. SMALL: In Wisconsin we are working along the lines of educating them to the use of mechanical ventilating equipment. They don't

realize the necessity of ventilation and in order to fool-proof the system we tie up the heating system in such a way that it can't heat without ventilating. We are not interested in whether it is a mechanical or gravity system or window system or so forth; we simply say in the code that you must have such and such results.

E. S. HALLETT: Down in St. Louis we probably have more ventilation than any place else in the world and we don't build anything but all blast systems. We have no law of any kind whatever to hamper us. We can demonstrate that you can ventilate and heat at the same time, and do it for less than with radiators and no ventilation.

F. R. STILL: We had a school in Minneapolis where we sent the air right into the faces of the pupils and they thrived on 10 cu. ft. Take any of this elaborate equipment, put it up as it is and you can't operate the school without operating fans. You can operate that Minneapolis school on half of what you can with any open window ventilation that does not come within 50 per cent of the desired results. If the public prefers the open window, let's determine definitely that it is the right thing to do. If we are fooling our time and the Society's on all this investigation, the sooner we know it the better. It is an economic waste if we do not determine it definitely.

W. H. CARRIER: We have had a basis of 30 cu. ft., which was established on the known fact that ventilation was an essential from practical experience. Just what amount was required was not known. I think that it is very probable that you can have 10 cu. ft. or 20 cu. ft. if you put it in the right way, but it is a question of distribution and not of amount. If you put in 30 cu. ft. and in such a manner that it just goes in and out, it doesn't mean anything and yet you are complying with the law.

J. W. H. MYRICK (written): I have found it good practice to make my fresh air inlet on the ceiling, a duct about 4 to 6 in. in depth and 14 to 16 in. in width with the proper number of small inch openings on the side to supply any air-change needed. This form of supply will act like a sprinkler system, allowing a well distributed amount of fresh air to fall like a plate into water and before it has worked its way down through the strata of overheated air at the ceiling, it has been tempered to a degree not objectionable, nor not over-heated and pasteurized and its freshness destroyed.

WALTER KEISER (written): Experiments indicate so far that mice, which have been poisoned more or less by their exhalation, are revived and brought back to their usual activity by ozone in concentrations estimated at about 1 part per million, and that if left in the air contaminated by their own exhalation they die. The action of the ozone seems to be the elimination of the poison of the exhalations by oxidation and when drawn into the lungs it tends to be selectively absorbed by the blood.

While the ozone concentration used here is considerably higher than that used in ventilation, according to Dr. F. O. Anderegg of Purdue University who conducted this experiment, the excess of ozone was needed under these conditions to take care of the exhalations.

No. 644

COMPARATIVE TESTS OF ROOF VENTILATORS

By F. B. Rowley[1], Minneapolis, Minn.

Member

UNTIL recent years very little attention seems to have been given to the comparative efficiency of different types of roof ventilators. Recently, however, several papers have appeared in the Journal which seem to indicate a growing interest in this field. It may, therefore, be of interest to report the following tests which, while not covering a complete range of ventilators, represent five distinct types, and these tests are therefore submitted as an additional contribution to the other data published.

The five different ventilators are illustrated in Figs. 1 to 10. The detail dimensioned drawings are given because it is apparent that the dimensions have much to do with the efficiency of the ventilator. It will be interesting to note that in tests of ventilator No. 1, the dimensions were changed, first by increasing and then decreasing the opening around the outlet. It was found that when this dimension was varied in either direction, the draft created by the ventilator was diminished, indicating that the proper dimensions had been selected in the original design. It appeared also that this dimension had been selected more by accident than by scientific determination.

The ventilators were all 10 in. in diameter and were mounted as shown in Fig. 11, which shows the general arrangement of the ventilator set up, together with the fan for producing the air velocity around the ventilator head. The wind velocity was produced by a No. 8, double-inlet Sirroco fan, discharging through a 36 in. circular duct. The ventilators were placed 3 ft. 10 in. in front of the outlet and in such position that the center line of the head was as near as possible in the center line of the duct. The ventilator pipe was supported in a frame, the bottom of the pipe being approximately 2 ft. from the floor line.

The wind velocity and the air velocity created in the ventilator shaft were measured with anemometers. In order to determine the wind velocity the ventilator pipe was removed and the wind velocity in the

[1] Professor of Mechanical Engineering, University of Minnesota.

Paper presented at the Annual Meeting of the American Society of Heating and Ventilating Engineers, Washington, D. C., January, 1923.

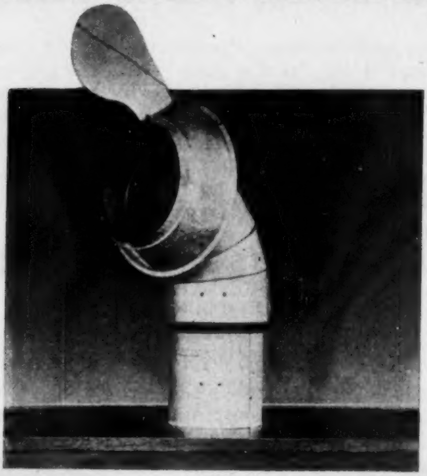


FIG. 1

FIG. 2

FIVE
STYLES OF
ROOF VENTILATORS
TESTED

FIG. 3

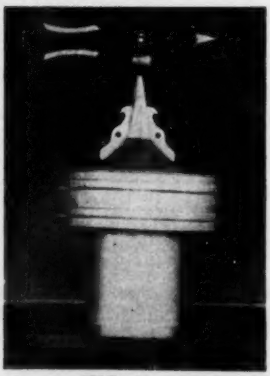


FIG. 4

FIG. 5

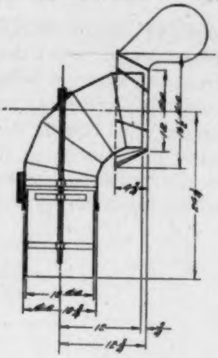


FIG. 6

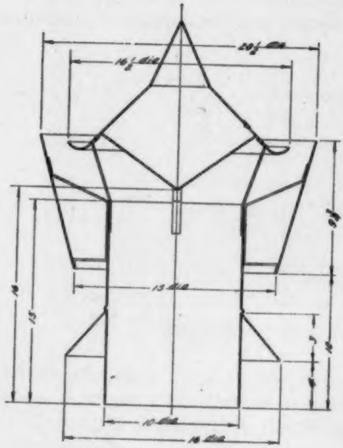


FIG. 7

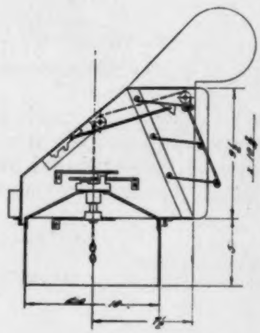


FIG. 8

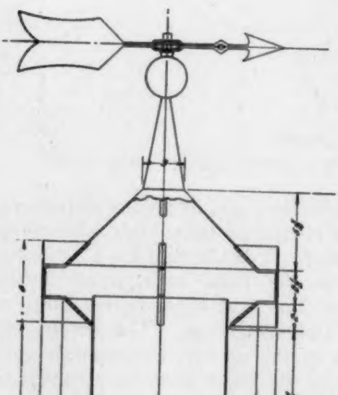


FIG. 9

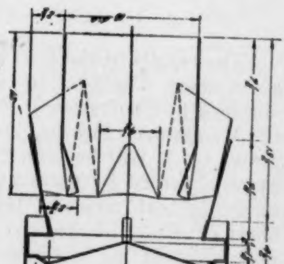


FIG. 10

plane of the ventilator taken at five different points, as shown in Fig. 12. The average was taken as the true wind velocity, the greatest variation from the average being less than 7 per cent. While this set-up may give somewhat different results than when the ventilator is placed in a wind tunnel, the results on the various ventilators will be comparative; and since the velocity of the air over the full section of the ventilator is practically uniform, as shown by the anemometer test, it should give results which approach practical conditions.

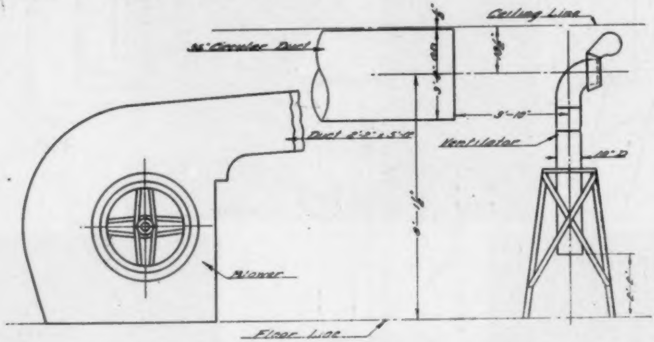


FIG. 11. VENTILATOR MOUNTED FOR TEST

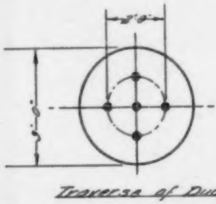


FIG. 12. TRAVERSE OF DUCT SHOWING ANEMOMETER READINGS

The results of the tests are shown in Table 1 and in the accompanying curve sheet, Fig. 13. In comparing the results of these tests with those obtained by Professors J. P. Calderwood, A. J. Mack, and C. J. Bradley of Manhattan, Kansas, and reported in the July, 1922, issue of the JOURNAL, it will be seen that ventilator No. 1 in these corresponds to those classified by them as the rotary siphoning type. The results obtained in both series of tests as shown by the curves, corresponds very closely. No. 4 of this test corresponds to the plane stationary ventilator and No. 5 of the stationary siphoning of the former tests. The results in these two cases are inverted, that is, the plane stationary giving greater efficiency in these tests, while the stationary siphoning gives the highest efficiency in the tests made at the Kansas Agricultural College. This may be accounted for by the different dimensions of the ventilators used.

Ventilator No. 2 in these tests would, no doubt, be classified in the former tests as plane stationary; yet this ventilator shows efficiencies

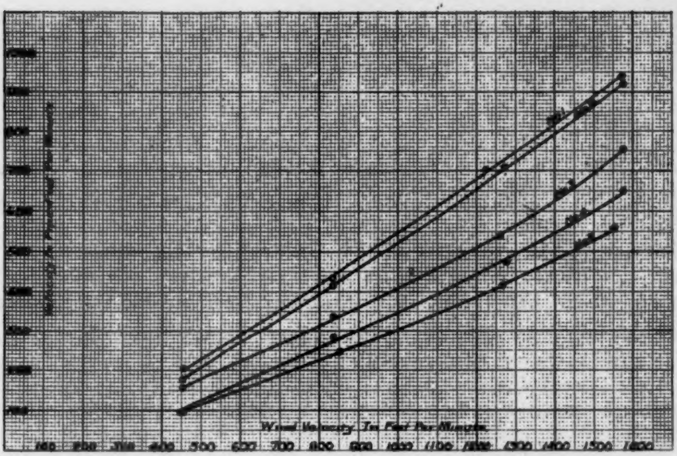


FIG. 13. CHART OF RESULTS OF ROOF VENTILATOR TESTS

which are very close to the best ventilators in either set of tests. The plane stationary ventilators in the former tests show the lowest efficiencies.

It would appear that it is impossible to take ventilators of the same class and expect uniform results, but that ventilators even though similar,

TABLE 1. TESTS OF ROOF VENTILATORS

Test No.	Ventilator	Average wind velocity	Average velocity in ventilator pipe
1	1	453	201
2	2	452	178
3	3	451	155
4	4	447	94
5	5	455	95
6	1	835	434
7	2	816	419
8	3	813	338
9	4	815	283
10	5	830	246
11	1	1225	702
12	2	1272	710
13	3	1238	539
14	4	1255	474
15	5	1244	417
16	1	1593	940
17	2	1572	919
18	3	1573	754
19	4	1572	650
20	5	1549	555

and classified as the same, will give entirely different results, depending upon the dimensions and proportions.

DISCUSSION

J. W. H. MYRICK: Would it be practical or better form to have an exhaust fan to free the air as drawn through the top?

H. L. DRYDEN: The ventilator was mounted outside the tunnel, just in front of the mouth, so that there was no static pressure difference in the system. Certain tests made elsewhere have been made inside the tunnel. In that case there is an additional pressure difference in the system. In our experiments the ventilators were outside the tunnel.

THORNTON LEWIS: Would discharging the air right into the room where the ventilator was, have any effect? Would it not be better if that pipe was connected to another room which is not in any way subject to pressure created by the fan forcing air into the tunnel?

H. L. DRYDEN: The velocity of the wind in the immediate neighborhood of the pipe line is very minute. The discharge is also very small, compared to the velocity of the wind

No. 645

HUMIDITY REQUIREMENTS FOR RESIDENCES

By A. P. Kratz[1], Urbana, Ill.

Non-Member

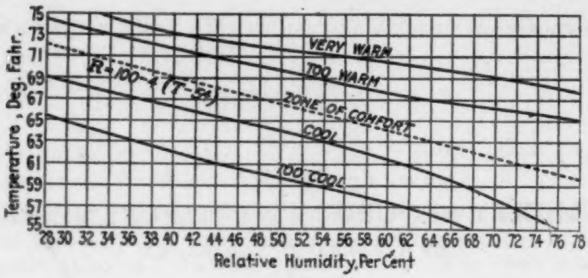
THE researches of Dr. E. Vernon Hill and other investigators have shown that the dry bulb temperature alone is not a reliable indication of the degree of comfort experienced in artificially heated residences, but that the relative humidity, or wet bulb temperature, plays an equally important part in giving the impression of warmth and comfort. At the same time the consensus of opinion among physicians seems to be that an atmosphere in which the humidity is abnormally low, tends to dry out the membranes of the nose, throat and lungs, thus throwing an extra load on these membranes and producing irritation, lowered vitality and a predisposition to colds and other respiratory troubles. Very dry air also takes the moisture from the wood-work of the house, the walls, and the glue in the furniture, thus causing warping, the appearance of unsightly cracks in both wood-work and plastering, and rapid deterioration of the furniture.

A publicity campaign on the part of several popular magazines has created a widespread interest in the problems relative to the degree of humidity necessary for physical well-being, the amount of water vapor required, and the means of evaporating water in sufficient quantities for adequate humidification. The data in this paper have been compiled for the purpose of answering, in some measure, the three problems stated.

The curves in Fig. 1 are a reproduction of some curves taken from the book entitled: The Mechanical Equipment of Buildings, Vol. I, by L. A. Harding and A. C. Willard, and are based on the results of ventilation tests made at the Chicago Normal College by the Chicago Commission on Ventilation, under the direction of Dr. E. Vernon Hill. Inspection of this chart will show that a room at 69 deg. fahr. will be comfortable if the relative humidity is maintained at a value of from 35 to 45 per cent. A relative humidity of about 40 per cent is as high as it is practical to maintain where residences are concerned. If the humidity is much higher than this, the condensation on the windows will be excessive unless double windows are used. An approximate rule is, that for single windows the numerical value of the relative humidity may be as

[1] Research Professor, Department of Mechanical Engineering, University of Illinois.

Paper presented at the Annual Meeting of the American Society of Heating and Ventilating Engineers, Washington, D. C., January, 1923.



R = 100—4(T-54).
R = Relative Humidity.
T = Any temperature above 55 deg.

NOTE: The above chart is based on the results of Ventilation Tests at the Chicago Normal College by the Chicago Commission on Ventilation under the direction of Dr. E. Vernon Hill. The data in the complete report show that a temperature of 64 deg. fahr. with 55 per cent relative humidity was as comfortable as a temperature of 70 deg. with only 30 per cent relative humidity.

FIG. 1. COMFORT CHART

high as the mean between the indoor and outdoor temperature before the condensation on the windows becomes excessive.

In Fig. 2 the mean monthly relative humidity for the past five years has been plotted against the mean monthly temperature. These data were taken from the observations made by the Department of Soil Physics at the University of Illinois, Urbana, Ill. Readings are taken at this Station daily at 7 a. m., 2 p. m. and 7 p. m. From the daily averages the mean for the month is obtained. This curve shows that while there is a tendency for the points to spread, the data can be fairly well represented by a mean straight line.

The amount of water in gallons per hour required to humidify 10,000

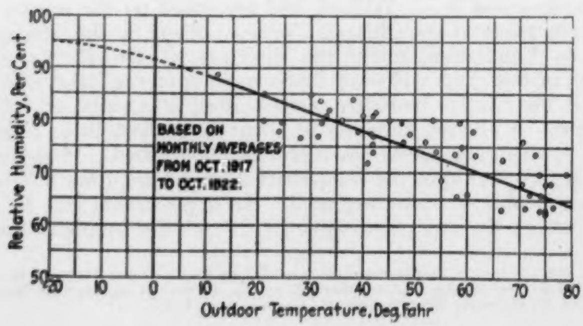


FIG. 2. RELATIVE HUMIDITY CURVE

cu. ft. of space may be found from the chart shown in Fig. 3. This chart has been calculated from the following formula:

$$G = \frac{VAd}{8.34}\left(\frac{r_1 w_1}{100} - \frac{r_0 w_0}{100}\right)$$

Where G = Gallons of water required per hour to humidify the house.
V = Volume of the space heated in cu. ft.
A = The number of air changes per hour caused by leakage of air from the outside.
d = Density of the air inside the house.
r_1 = Relative humidity indoors in per cent.
r_0 = Relative humidity outdoors in per cent.
w_1 = Weight of water vapor in one lb. of saturated air at the temperature indoors.
w_0 = Weight of water vapor in one lb. of saturated air at the temperature outdoors.

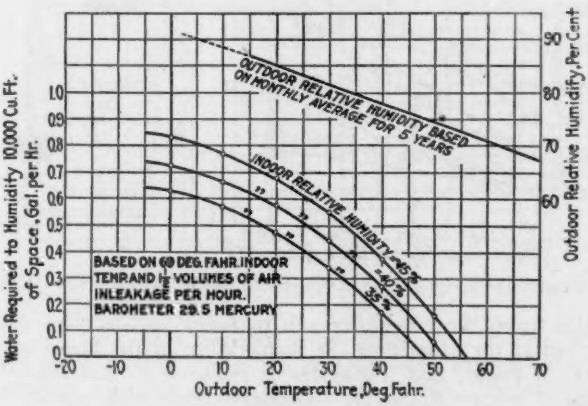


FIG. 3. HUMIDIFICATION CHART

In calculating the chart in Fig. 3 the value for the relative humidity outdoors, r_0, was taken from the curve in Fig. 2, which has also been transferred to Fig. 3. It was assumed that the temperature maintained in the house was 69 deg. fahr., and that the number of air changes per hour due to inleakage was 1½. The values of w_1 and w_0 were taken from the table of mixtures of air and saturated water vapor given in Goodenough's Properties of Steam and Ammonia. The volume, V, was taken as 10,000 cu. ft. When these values are substituted, the formula becomes:

$$G = 1.35\ (r_1 w_1 - r_0 w_0)$$

For example, for a relative humidity indoors of 40 per cent and a temperature outdoors of 20 deg. fahr.:

$$G = 1.35\ (40.0 \times 0.01524 - 85.0 \times 0.00214) = 0.578$$

In order to use the chart, a reasonable value is first chosen for the minimum outdoor temperature. This should be the same as the outdoor temperature upon which the design of the heating system is based. By reference to the chart in Fig. 1, the proper relative humidity indoors may be determined. The amount of water required to be evaporated per hour may then be read from Fig. 3. This gives the humidity required for the daytime, during which the temperature to be maintained is 69 deg. fahr., and is the maximum demand for the evaporative performance of the humidifier. At night, when the heating system is less active, the

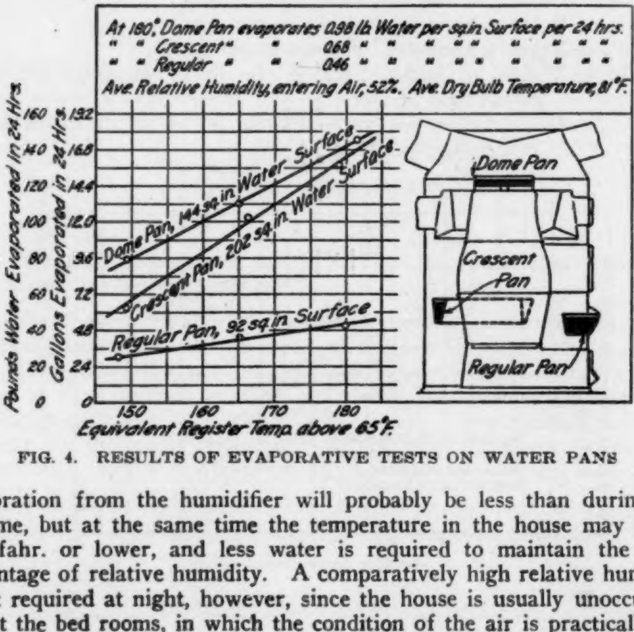


FIG. 4. RESULTS OF EVAPORATIVE TESTS ON WATER PANS

evaporation from the humidifier will probably be less than during the daytime, but at the same time the temperature in the house may be 60 deg. fahr. or lower, and less water is required to maintain the same percentage of relative humidity. A comparatively high relative humidity is not required at night, however, since the house is usually unoccupied, except the bed rooms, in which the condition of the air is practically the same as outdoor air if they are properly ventilated. If the humidifier has a capacity large enough for the day time requirements it will, therefore, prove sufficient for those at night.

In the case of a warm-air furnace system, when the outdoor temperature increases, the temperature at the register faces decreases. The evaporation will, therefore, become less, and tend to compensate for the fact that less water is required when the outdoor temperature is higher.

For example, in order to humidify a house of 15,000 cu. ft. capacity so that it is comfortable at 69 deg. fahr., with an outdoor temperature of zero deg. fahr., the indoor relative humidity should be maintained at about 40 per cent. Reference to Fig. 3 will show that an evaporation of $1.5 \times 0.725 = 1.09$ gal. of water per hour will be required of the humidifier during the day time.

The curves in Fig. 4 show the evaporative performance of three types of water pans for a warm-air furnace fired with anthracite coal. These curves were obtained as incidental to a co-operative investigation of warm-air furnaces conducted by the Engineering Experiment Station at the University of Illinois in co-operation with the *National Warm Air Heating and Ventilating Association,* and were previously published in the JOURNAL of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS for January, 1922. From these curves it is apparent that the evaporative rate for the pan on the dome of the furnace is about

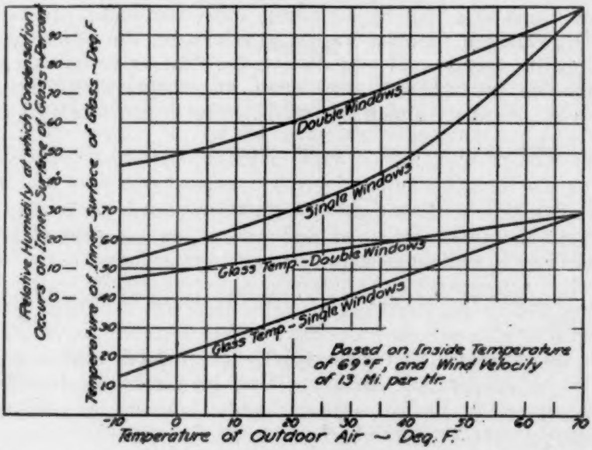


FIG. 5. CONDENSATION CURVES FOR GLASS WINDOWS

0.71 gal. per hour at a register temperature of 180 deg. fahr. and, therefore, more nearly meets the requirements for humidification in severe weather than any of the other types shown. If bituminous coal is used the evaporative rates will be less than those shown in Fig. 4.

The curves in Fig. 5 show the temperature of the inside surface of the glass, and the relative humidity at which condensation will appear on the inside surface of the windows, for an indoor temperature of 69 deg. fahr., and varying outdoor temperatures. These curves indicate that when single windows are used, with a relative humidity of 40 per cent, condensation will appear on the glass when the outdoor temperature is approximately 30 deg. fahr. If storm windows are used, the outdoor temperature may be as low as —10 deg. fahr., without condensation occurring, even with a relative humidity of 40 per cent.

DISCUSSION

H. L. WILLIAMS: The test in this paper relates to recirculating systems only, and of course the possibility of maintaining 40 per cent

saturation, or evaporating the amount of water stated, is increased to a very considerable extent because the humidity of the house itself is given to that air. But if you were to make the attempt to maintain that same humidity by taking your air from the outside fresh you would soon fail. We used humidity pans for quite a number of years, but the difficulty, of course, is in keeping them filled. We have always used a float valve wherever it could be connected up with the water system.

This paper for the purpose of recirculated air is alright. But when you come to taking air from the outside and keep constantly changing that air as you do with a specially ventilated system, it does not hold. We have found also that by no means must the water pan be put on the inside; although the evaporating pans can be, the filling pans must be kept on the outside. If you do not, the float valve will soon be out of commission because sand and other materials will get into it. It must also be put where a man can see it, get at it and take care of it.

THE PRESIDENT: What is your experience as to keeping the pans sanitary?

H. L. WILLIAMS: It is most difficult when put on the inside, as scum will gather on top of the water and where you are dependent on the human element it will not be kept in a very sanitary condition.

THE PRESIDENT: Some believe that the depth of water has a lot to do with that.

H. L. WILLIAMS: Our pans are 10 in. in depth and each one has 540 sq. in of evaporating surface. You get better results where you can feed your water from the outside, and I have ceased putting in any filling pans of that kind unless they are on the outside.

R. A. SMALL: In Wisconsin we are more interested in results than in methods, and we do not approve a water pan that is manifestly inadequate for the job intended. Our problem is to get a water pan in a place and of a size that will deliver moisture in the air. As a rule, we find that the better performance comes from the dome pan. Part of the trouble comes from the fact that they like to put the dome pan on the side of the heat chamber where it does not interfere with the valves into the flues, but neither does it treat all of the air handled as it should.

H. L. WILLIAMS: It has been represented that you can maintain 40 per cent of saturation with an ordinary pan handling a great quantity of air. It cannot be done. A high absolute temperature is all right, but where you have a high absolute temperature you get too much as a rule and where you are starting with a low absolute humidity, you will not get enough.

No. 646

HEATING THE NEW NAVY GUN SHOP

By M. S. Cooley[1], Washington, D. C.

Member

THE new gun shop at the Washington Navy Yard is the largest of six new buildings located in the west extension to the yard. These buildings form a part of the enlargement program being carried out to increase the size of the principal Navy Yards. This plan was started in connection with the increased naval building program authorized by Congress and carried out by the Bureau of Yards and Docks in 1916.

Owing to the great number of large buildings contemplated, the bureau required that the design for the steel work be submitted with the bids and that details for steel work be prepared by the successful bidder. Before these details for the steel work were recived the United States had entered the World War, entailing rush work on overseas structures and preventing a study of the heating system until after designs for the steel work had been completed by the contractors.

The heating system was designed in the Bureau of Yards and Docks. Navy Department under direction of Admiral F. R. Harris, chief of the bureau, after details of steel work had been completed by the contractors, and it was then found that the clearances between columns were just sufficient to allow the largest guns to be passed when swung from cranes. Therefore no longitudinal air ducts could run in main aisles, and the cross aisles containing the gun lathes could not have ducts projecting into them, without danger of their being damaged in event that a gun should swing sideways, striking the columns. No space was available on the floor for heating units and no room between foundations for underground ducts in about 80 per cent of the floor area, so that the heating-system design was quite a serious problem.

Owing to the difficulties above stated, the officials at the naval gun factory recommended that the building be heated by direct radiation. It was estimated by the Bureau of Yards and Docks that 67,000 sq. ft. of radiation would be required, and a study of the wall space available indicated that it was impossible to install this amount of radiation in the space provided, since the installation of pipe trenches had to be limited. This

[1] Bureau of Yards and Docks, Navy Department.

Paper presented at the Annual Meeting of the American Society of Heating and Ventilating Engineers, Washington, D. C., January, 1923.

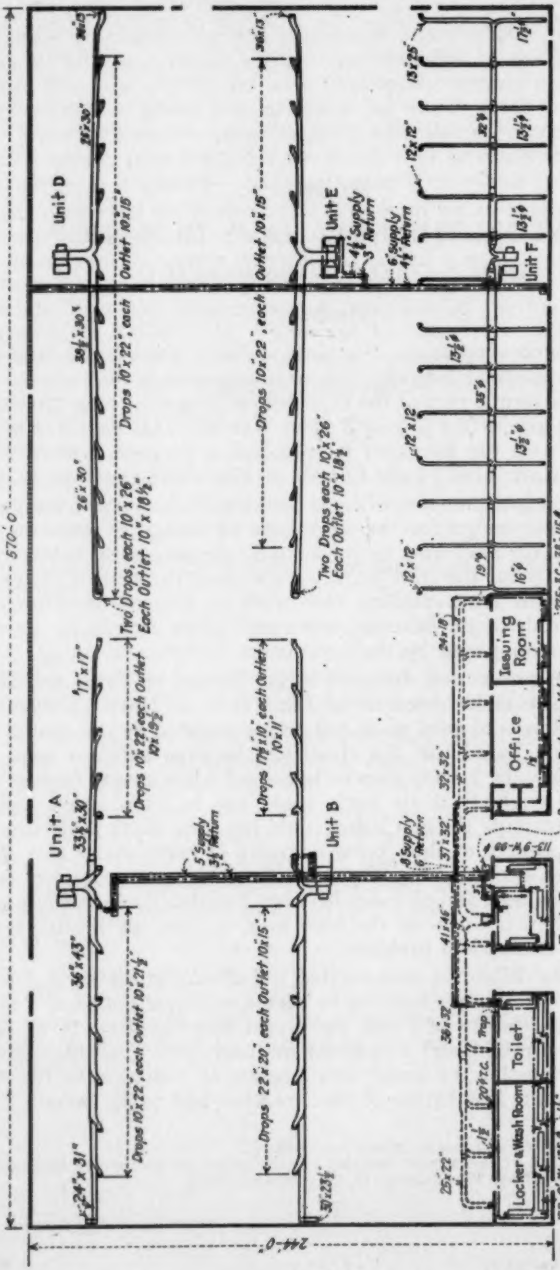


FIG. 1. LAYOUT OF DUCTS FOR HOT BLAST HEATING SYSTEM IN NEW NAVY GUN SHOP

system was therefore considered impracticable, and it was decided to heat the building by the fan blast method.

The problem therefore was to heat a building 570 ft. long, 244 ft. wide, varying from 40 to 135 ft. in height, by fan blast system, using recirculated air, with the heating units from 30 to 72 ft. above the floor and with no space available for recirculating ducts and where no distributing ducts could be run in aisles in either direction.

This problem was solved by locating six heating units in the roof spaces above the crane clearances, running distributing ducts in this same space and bringing the outlet branches down close to the columns on such

FIG. 2. VIEW IN MAIN AISLE OF NAVY GUN SHOP SHOWING DUCT WORK BESIDE COLUMNS

sides so that they would not interfere with gun clearances or render them liable to damage by being hit by a swinging gun. This system is entirely satisfactory and does not occupy any of the space necessary for use in the manufacture of the guns, and the installation cost was about 40 per cent of that of a system using direct radiation.

The distribution system for the west yard was laid out to eventually use forced circulation hot water with a pressure difference of 15 lb. inside each building, but to be temporarily operated with steam, using 5 lb. pressure at the entrance to the building. It was therefore necessary to so proportion the supply and return piping that the friction in the two lines would be sufficient to produce a pressure drop of 15 lb. at the entrance to the building, when the quantity of water required for heating the building was passing, and at the same time keep the supply pipe of sufficient size to pass the required amount of steam with a friction loss

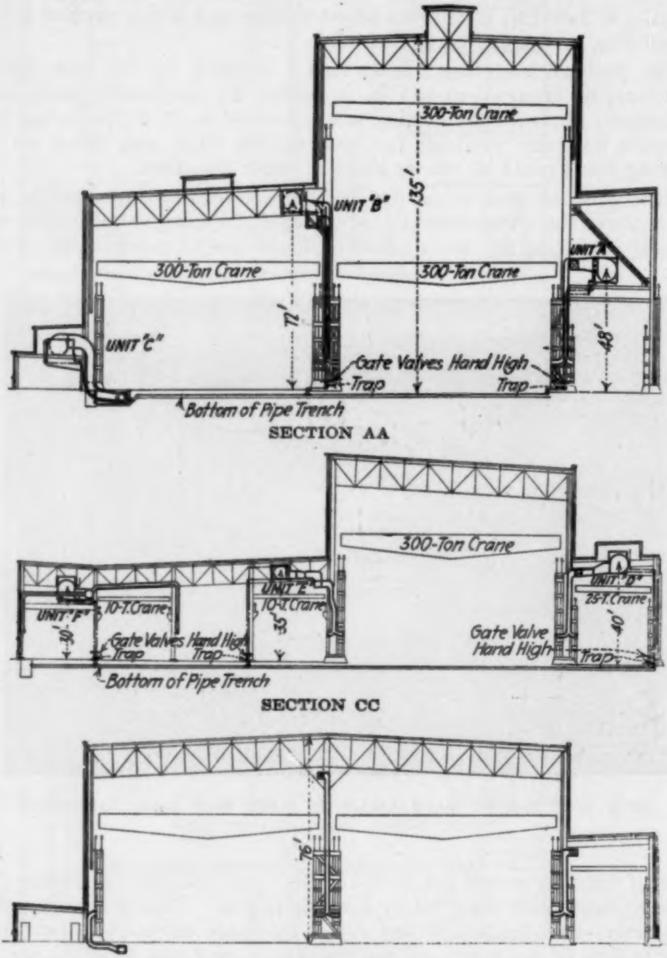


FIG. 3. SECTIONAL VIEWS SHOWING VENTILATING UNITS IN NEW NAVY GUN SHOP

of less than 5 lb. Traps were temporarily installed on the return from each heater and direct radiator.

The total exposure of the building is:

Glass	89,995 sq. ft.
Net terra cotta and concrete walls	75,707 sq. ft.
Concrete roof	127,869 sq. ft.
Concrete floor on earth	127,000 sq. ft.

The total heat loss was estimated at 16,735,000 B.t.u. heating building from 10 deg. to 60 deg. fahr.

The total capacity of the six fan units is 223,000 cu. ft. per min. and the total vento heating surface 15,495 sq. ft. heating air from 50 deg. to 140 deg. fahr.

The total cubic contents of the building is 8,995,000 cu. ft. and the fans installed give a 40 min. air change. This air change would be considered inadequate by many of the leading manufacturers of fan blast heating systems, but, it was found to be impossible to install larger apparatus in the space available.

This lack of space for larger apparatus and the impossibility of in-

FIG. 4. VIEW IN WING SHOWING LARGE DUCT FROM HEATING UNIT E AND OFFSET DROPS

stalling recirculating ducts from near floor to fan inlets was overcome by the use of high velocities at air outlets and careful design of outlets to give as near a uniform air distribution as possible and the results have been satisfactory. The outlet velocity varies from 1000 to 1300 ft. per min., with all outlets 10 ft. above floor.

The gun shop is a steel frame structure enclosed with 6 in. terra cotta walls, reinforced concrete roof, and steel sash. Over 50 per cent of the walls of this building, which is 570 ft. long, 244 ft. wide, is glass. The northwest section, 135 ft. high, contains a pit 100 ft. deep in which are located electric furnaces for heating jackets into which the gun liners are lowered and jackets shrunk on. This portion of the building contains a 300-ton crane with 100 ft. clearance under the hook. Adjoining this on the south is a section 76 ft. high, in which the breech mechanism is fitted

to the guns and also contains lathes for turning jackets. This section is served by a 300-ton crane with 40 ft. clearance under the hook.

Directly east of the space containing the shrinkage pit is a building serving as housing for the third 300-ton crane which passes the guns from the shrinkage pit to the boring and turning lathes and rifling machines; this section is 76 ft. high and the travel of the third crane is 570 ft.

To the south of the last named space is a 40 ft. high section which houses the bed plates and boring bars of the gun lathes with two 10-ton cranes for handling the lathes and their attachments. North of the section which is 76 ft. high, is a lean-to, housing the head stocks and driving motors of the gun lathes with one 20-ton crane for handling machinery.

There are six fan and heater units to produce and distribute heat for the building. Unit *A*, with a capacity of 48,000 cu. ft. per min., is located on a platform over the north lean-to and distributes air through a horizontal duct system running along the same platform with ducts dropping down alongside of columns to a point 10 ft. above the floor, care being taken to keep ducts out of bays containing railroad tracks. The intake for this fan is 48 ft. above floor.

Unit *B* has a capacity of 35,000 cu. ft. per min. and is located in the roof trusses between the northwest and southwest portions of the building. Air is distributed through a horizontal duct placed in the roof trusses. Since the inlet of this fan was 72 ft. above floor, no attempt was made to recirculate this air, the fresh air being admitted through a clerestory window.

Unit *C*, having a capacity of 35,000 cu. ft. per min., is located on the roof of the locker room and distributes air through a system of underground ducts, which were possible as there are no heavy machinery foundations in the space reserved for fitting breech mechanisms.

Unit *D*, rated at 35,000 cu. ft. per min., is located partly in trusses over the north lean-to and partly in a housing above roof and distributes air through a horizontal duct located in the trusses over the north lean-to, with ducts dropping down columns. The inlet to this fan is 40 ft. above the floor.

Unit *E*, with a capacity of 35,000 cu. ft. per min., is located in roof trusses over the 40 ft. high section, where gun turning lathes are housed, its inlet being 35 ft. above the floor and distributing its air through a horizontal duct located in roof trusses of this section.

Unit *F*, capacity 35,000 cu. ft. per min., is also located in the roof trusses of the 40 ft. high section and distributes air through a system of ducts located in roof trusses, the drops from which are offset around the columns to prevent damage by a swinging gun or boring bar. This is the only place in the building where there was sufficient space available to permit the use of round ducts.

The photographs, Figs. 1 and 4, show the main aisle looking toward the shrinkage pit, giving an idea of the duct work on the large crane columns, and the under side of Unit *E* giving an idea of the offset drops in the gun lathe section.

The toilets, locker rooms, office, tool and issue room in the south lean-to are heated by 2,000 sq. ft. of direct radiation. All fans are motor driven, remote push-button control for stopping and starting motors from the floor of the shop are provided. Volume dampers are installed in all outlets and deflectors in main ducts; no temperature control or speed control of fans is provided. Hand control of the vento sections is relied upon for controlling the temperature of the room, which was figured from + 10 deg. fahr. to 60 deg. fahr.

The heating apparatus was installed by Mathis Bros. of Chicago, Ill., under supervision of the Public Work Department, Washington Navy Yard.

DISCUSSION

W. H. CARRIER: In a very interesting application Mr. Cooley points out the advantages of high velocity in getting the mixture. My only criticism is he did not go quite high enough because I believe that the best velocity for the ordinary building of that structure, where you cannot blow down is 1500 or above. You do not need high velocities in your ducts unless you are forced to get them on account of space. Keep your velocities in your duct down, proportioning them in the usual way as you would for a lower velocity job and then decrease, if anything, the area of your outlet making them smaller than the area of your pipe. It will run your stack pressure up, but in high buildings I like to see that 2000. Your mixture is directly proportional to your velocity and it is mixture you want. You don't want high velocity when you want a circulation in the building. You want a velocity that would not be objectionable for drying work.

M. S. COOLEY: In this case we were limited to high velocities in main ducts. We put as large a duct as we could and it ran about 3000 ft. The circulation inlet was not on the floor but in the air. The temperature was 140 deg. rated in the heater and we figured to get 130 at the outlet. The building was heated to 60 deg. and in +10 deg. weather we had a 50 deg. rise in the building. They wanted this system laid out for forced water and still to use 5 lb. of steam. We designed it for forced water at 15 lb. differential at the building and yet made it work on 5 lb. steam at that point. It was necessary to make the two sides of the line, which was to be hot water, one bigger than the other because we had to make the supply pipe big enough for steam, generally two sizes larger than the return pipe, but we used 5 lb. pressure. Condensate from heater was individually trapped out and came back in the return under atmospheric pressure. There was no vacuum pump.

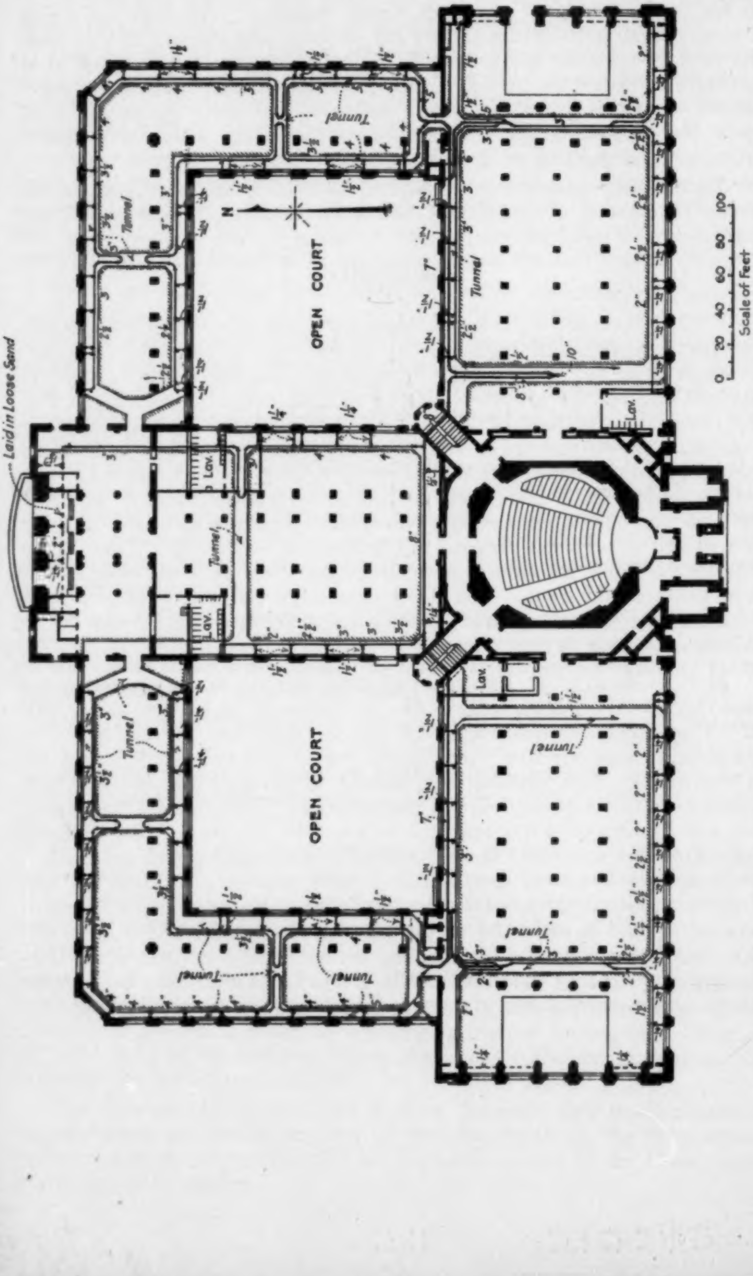


FIG. 1. BASEMENT PLAN OF U. S. MUSEUM SHOWING SIZE AND RUN OF HOT WATER HEATING SYSTEM PIPING

No. 647

HEATING AND VENTILATING U. S. MUSEUM

By C. R. Denmark[1], Washington, D. C.

Non-Member

THE Natural History Building, Washington, D. C., better known as the new building of the National Museum, is located in the Smithsonian grounds immediately north of the Smithsonian Building, the north and south axis of both buildings corresponding with that of Tenth Street extended, the two being separated by a distance of about 725 ft. The Museum faces north on B Street, with its east and west axis parallel with the axis of the mall, a line drawn from the dome of the Capitol to the Washington Monument. The building along the south front extends 561 ft. and along the north 499 ft. 4 in., while along the east and west it is 313 ft. 2 in. in length. There are projections at the north and south fronts, which, added to the last dimension just given, makes 364 ft. 6 in. along the north and south axis of the building. There are two large uncovered courts, 120 ft. 2 in. square, formed by the wings and ranges. The main entrance, facing the mall, gives direct access to the rotunda on the first floor, while a second entrance in the center of the opposite or north front, at the basement or ground floor level, gives direct entrance from B Street at a point not far distant from Pennsylvania Avenue, the main route of travel in the National Capital.

The building was planned so as to better meet the requirements for exhibition purposes with but few permanent dividing walls, in this way providing unusually large floor area as compared with the outer wall. Another noticeable feature of the structure is the large glass surface, as compared with the wall surface, which is unusual for a monumental type of building.

It is four stories in height, exposed and finished on all sides and occupying a ground area of nearly four acres. In plan it consists of a large pavilion in conjunction with a rotunda facing south, from which there radiates three wings—one to the north, one on the east and the third to the west, these wings in turn being connected at their outer ends by L shaped ranges so as to form two completely enclosed courts. This gives the building a rectangular shape when viewed from the outside, with the

[1] Engr., Smithsonian Institution, U. S. National Museum.

Paper presented at the Annual Meeting of the American Society of Heating and Ventilating Engineers, Washington, D. C., January, 1923.

east and west halves exact duplicates. The general appearance is that of a monumental structure of brick and stone, the outer walls being faced with gray and white granite, the court walls with gray and white brick

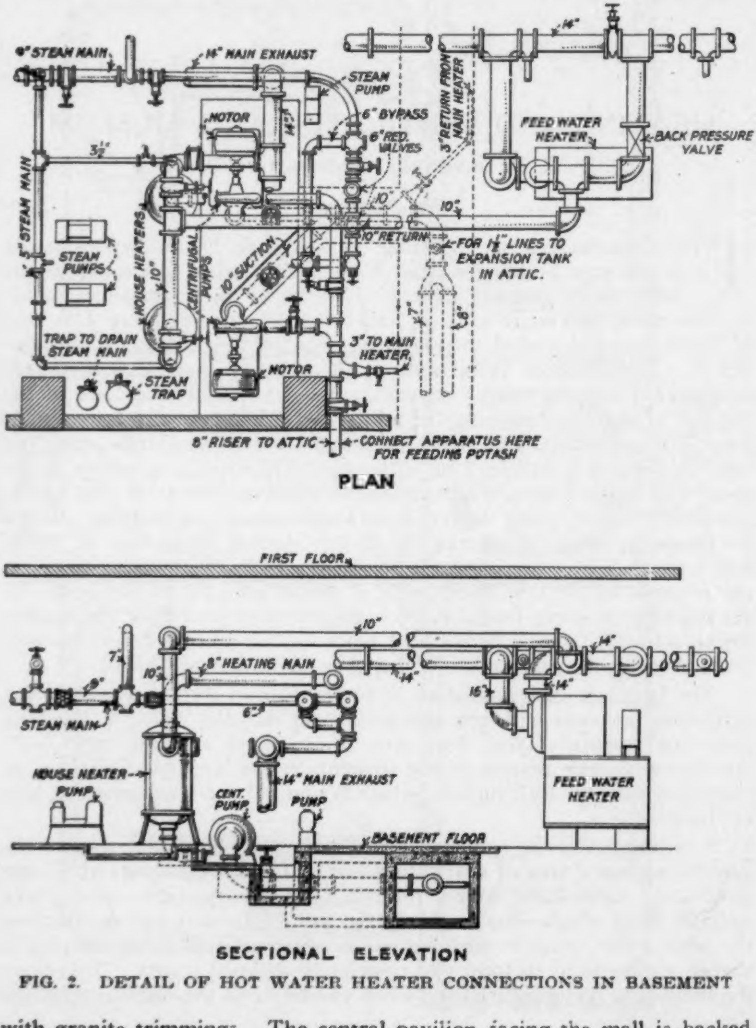


FIG. 2. DETAIL OF HOT WATER HEATER CONNECTIONS IN BASEMENT

with granite trimmings. The central pavilion facing the mall is backed by a dome 74 ft. 6 in. in diameter at the base, which rises to a height of 162 ft. from the basement floor, thus forming the commanding feature of the structure. Around the wall of the dome are located four semi-

circular windows rising from the fourth floor level to the base of the dome, for providing light for the main portion of the rotunda itself.

In order to gain a definite idea of the size of the building, it is well to consider certain dimensions such as the cubic contents and outside dimensions of the building itself. The total cubic contents within the four walls is 11,604,984 cu. ft., which covers a total ground area, including the two closed courts of 168,752 sq. ft., with the building actually covering 136,171 sq. ft. The total floor area within the walls including the four main floors as well as the available floor space in the attic, is 468,118 sq. ft., distributed as follows: basement, 121,173 sq. ft.; first floor. 119,480 sq. ft.; second floor, 86,357 sq. ft.; third floor, 86,428 sq. ft.; attic, 54,680 sq. ft.

Of the floor area, more than one-half the total surface is given over to the use of the public, as exhibition space, entrance halls or auditorium. the remainder being used as offices, workshops, laboratories, engine and boiler rooms and storage space.

As previously stated, the design of the building is such that the general dimensions of various portions are identical in different parts of the building. The typical dimensions of each bay, that is, the distance between the piers separating the windows, is 18 ft. 6 in. With but few exceptions, therefore, the dimensions of the window openings and skylight areas are such that the glass surface given in one typical window, multiplied by the number of windows, will give the total glass surface in all of the building.

Approximately three-fourths of the area of each of the windows on the ground floor is actually glazed, the remaining one-fourth being taken up by window frames, mullions, etc. On the court walls there are 50 windows, which, added to the small windows occurring in the stairwells and towers, give a total window area of 6850 sq. ft., while in the outside walls there are 73 windows and glazed doors, which, added to the several smaller windows occurring on the south and north pavilions, give a total area of 7750 sq. ft. This makes a total of 14,600 sq. ft. for the basement floor.

In the first and second floors, where a major portion of the exhibits are placed, the windows take up a larger portion of the surface. The window openings are continuous through the two stories, being 31 ft. 10 in. high in the outer walls in the east and west wings, and 31 ft. 6 in. high in the courts and outer walls of the ranges, from the top of the masonry sill, which is 3 ft. 11 in. above the level of the first floor. The total window openings on these two floors amount to 45,480 sq. ft., but as there is a solid panel 3 ft. wide dividing the windows between the first and second floors, there should be deducted 4100 sq. ft. from this amount, leaving a final total 41,380 sq. ft. As these windows are built with metal frames, there is a very large percentage of the openings actually glazed.

On the third floor the window openings are much less in area than those on the first and second floors, made with the usual movable sash. In the east and west wings, along the south side and outer end of each

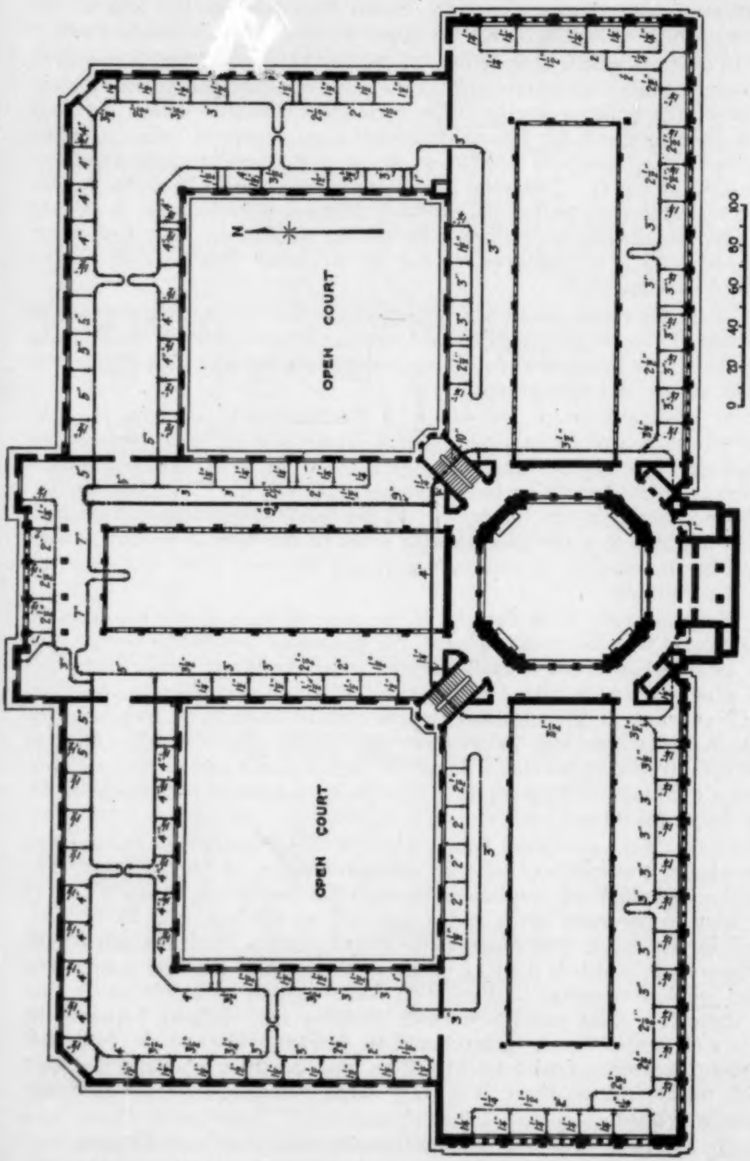


FIG. 3. FLOOR PLAN SHOWING RUN AND SIZE OF HOT WATER LINES IN ATTIC

wing, there are 64 window openings, there being two to each bay, to which should be added 10 windows of approximately the same size along the front of the north pavilion. The window openings in the dormers of the ranges, both east and west are 38 in number, while there are 50 window openings in the two courts. This gives a total window surface on the third floor of 7180 sq. ft.

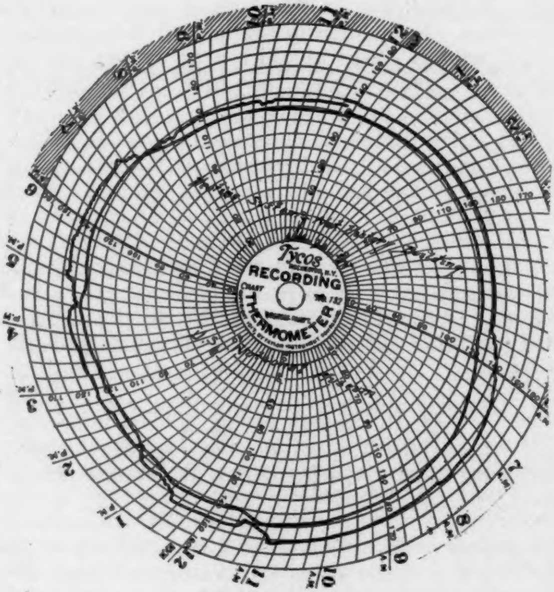


FIG. 4. RECORD SHEET WATER TEMPERATURE

There are three main skylights in each of the three wings, those in the east and west being 32 ft. 2 in. wide by 149 ft. 4 in. long, while the one in the north is 32 ft. 2 in. wide by 149 ft. 10½ in. long, giving a total skylight surface in the three wings of 14,395 sq. ft.

As previously mentioned, there are four semi-circular dormer windows around the rotunda, three of which are in line with the three wings while the fourth faces the south and is directly above the south portico. These windows are approximately 20 ft, in radius and with total glass area of 2550 sq. ft. In addition to these is a central skylight in the eye of the dome 18 ft. 10 in. in diameter.

There is then a grand total of exposed window and skylight area of 80,110 sq. ft., with more than half of it on the two main or exhibition floors.

HEATING

In order to heat the structure already described, the method adopted was hot-water with forced circulation, the force of circulation being provided by volute pumps placed in the engine room of the building, which are under the direct control of the engineer on duty. In order to provide sufficient heating surface for the different portions of the building at the maximum water temperature of 75 deg. in zero weather, there has been installed a total of 73,385 sq. ft. of direct radiation made up of cast iron sections. In addition to the total of direct radiation, there is 2480 sq. ft. of indirect, a part of which is for heating and ventilating the auditorium and a small portion for heating the upper part of the rotunda.

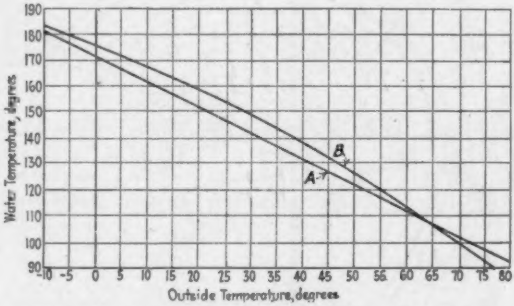


FIG. 5. CHART SHOWING CALCULATED WATER TEMPERATURE, *A*, REQUIRED FOR HEATING BUILDING AND ACTUAL TEMPERATURES, *B*, FOUND IN OPERATION OF PLANT

The large glass surfaces occurring in the windows and great volume of the halls made it necessary to install large and long radiators, some of which are 50 sections in length, with heating surfaces of more than 200 sq. ft. They are two and three column, varying in height from 22 in. to 38 in., made up with screwed nipples.

The method provided for circulation of water throughout the building was a one-pipe system. The unusual feature in the system lies in the manner in which the radiators are connected to a single pipe, there being two 1½ in. connections from the single drop, both of which are made at the same end of the radiator. There are 130 of these drops, in fact they are universally that size, except along the south side where they are 1¼ in. in diameter. From this one drop the two connections are made to the radiators, the supply at the tops, with the return at the bottom, both at the same end. In order to provide sufficient water to each radiator, long turn fittings were used and the connections to the drops made in such a manner as to assist the natural flow through the radiator, the supply coming from the drop to the top of the radiator from a T. Y. looking down, while the return to the drop from the radiator is made by a T. Y. looking in the opposite direction. This method of connecting the radia-

tors insures positive circulation to each radiator and it has been found in practice that this arrangement gives positive heat to each radiator throughout its entire length, even though circulation of water through the system may be at a minimum.

The typical drop has connection to four radiators, one on each floor, the total heating surface ranging between a minimum of 325 sq. ft. to a maximum of 807 sq. ft. Each drop is controlled by a cut-off valve in the attic and another in the tunnel, with a drain cock in the latter and an additional lock-shield valve in the former for individual control of flow to the separate drops.

The system is a simple attic distribution of hot water with drops for furnishing the required amount of the heating medium to the individual radiators. From the heaters in the engine room an 8 in. main rises directly to the attic from which there radiates the mains for distributing to all parts of the building. The velocity of flow in the riser was allowed to be approximately twice as great as in the distributing mains, hence the great increase in size. As the return mains in the tunnels were allowed the same, drop, they have the same area as the attic mains. A good idea of the system may be obtained from Figs. 1 and 3.

For removing air, there are automatic air traps, properly located in the attic mains, so as to remove the air that collects and retards the flow of water. As these mains, for structural reasons, could not be properly graded, more of these air traps were used than would ordinarily be found necessary.

A small one-story building in the East Court, 68 ft. 5 in. x 57 ft. 11½ in. is heated from the system in the main building. A 2½ in. line branches from the 3 in. supply connection to the heaters in the fan room of the auditorium, passes down through a trench to the East Court building, where it makes a complete circuit returning through the same trench and connecting to the 3 in. return from the heaters in the fan room. All of the branch connections to the radiators are 1¼ in. and long-turn fittings are used throughout, together with long-turn bends. The total heating surface in this building is 1265 sq. ft. consisting of three radiators on the south, three on the north, and one each on the east and west walls.

The plant for heating the water in the system is located in the engine room in the basement of the east wing of the building and consists of two complete units. Each unit is made of an Alberger condenser, with cast-iron body and corrugated copper tubes expanded into cast-iron heads, so designed that the water makes one passage through the heater, passing within the tubes, which are surrounded by steam space contained within the cast-iron body. There are two connections to the cast-iron body, one on the side near the top for the admission of steam, and one on the side near the bottom for the removal of condensation, while there are two 8 in. connections to the tube space, one at the bottom connected with pump discharge and another at the top for discharge to heating system. The water in the heating system returns through the mains in the tunnels finally reaching an 8 in. Alberger volute pump, directly connected to

and driven by a 20 hp. Westinghouse motor. The speed of the pump can be regulated so as to pass a minimum of 70,000 gal. of water an hour and a maximum of 110,000 gal. an hour. This speed regulation was introduced in order to maintain the constant temperature difference between the incoming and outgoing water to the heater. The steam supply to the heater is connected with the main exhaust line in the engine-room by means of a 10 in. pipe with control valve and also with the high pressure steam line by means of a 4 in. pipe with control valve. The exhaust line reaches the heater after passage of all exhaust steam through an open feed water heater. The details of this water heating plan are clearly indicated in Fig. 2.

The manipulation of the several valves and the speed of the pump gives absolute control of the temperature in the building to the operator in the engine-room. The temperature of the discharge water from the heater is determined by the outside temperature of the atmosphere, through the medium of a temperature chart. This chart was originally determined to be a straight line, but has since been found incorrect for this particular building. This chart is based on a temperature difference of 10 deg. between the supply and heater return. A curve has been established and is the one by which the operator regulates the temperature of the discharge from the heater. During the time the building is closed to the public or is not occupied, only sufficient live steam is used to maintain the temperature to within 20 deg. of that called for on the chart. In the event that there is more exhaust steam produced than necessary to heat the water to the proper temperature, the operator decreases the amount fed to the heater by means of a valve adjacent to the heater itself, which automatically raises the temperature in the boiler feed water heater until a maximum of 212 deg. is reached; a slight pressure is then built in the main exhaust line which is automatically relieved by means of a back pressure valve, allowing any excess steam to pass to the atmosphere. In the event that there is not enough exhaust steam to raise the temperature of the water to the desired point, the high pressure valve is opened sufficiently to accomplish this, while in the event the temperature difference of the heater supply and discharge is greater or less than 10 deg., the speed of the pump is changed accordingly.

The condensation from the heater is removed by means of a steam driven pump, which tends to create a slight vacuum in the heater itself, insuring at all times a complete drainage from the heater in spite of the fact that there might be a vacuum formed, due to a low temperature of the water in the heating system. If an effort were made to remove this condensation by means of a steam trap, there would be considerable difficulty experienced when the temperature of the water in the heating system is low, as at that time there would be a tendency for the heater to act as a condenser, thereby causing a vacuum and preventing the flow of water to the trap by gravity.

In addition to the 4 in. steam connection mentioned above, there was provided a direct high-pressure steam connection through an automatic reducing valve. This connection has not, however, worked satis-

factorily, due to the fact that the pressure in the heater varies from time to time and at no time is it sufficiently great to cause the regulation of the flow of steam to be controlled by a reducing valve.

A continuous record of the temperature of supply to, and discharge from the heater is made by a recording thermometer, the pens making the record using two different colored inks, produces a chart, Fig. 4, easily

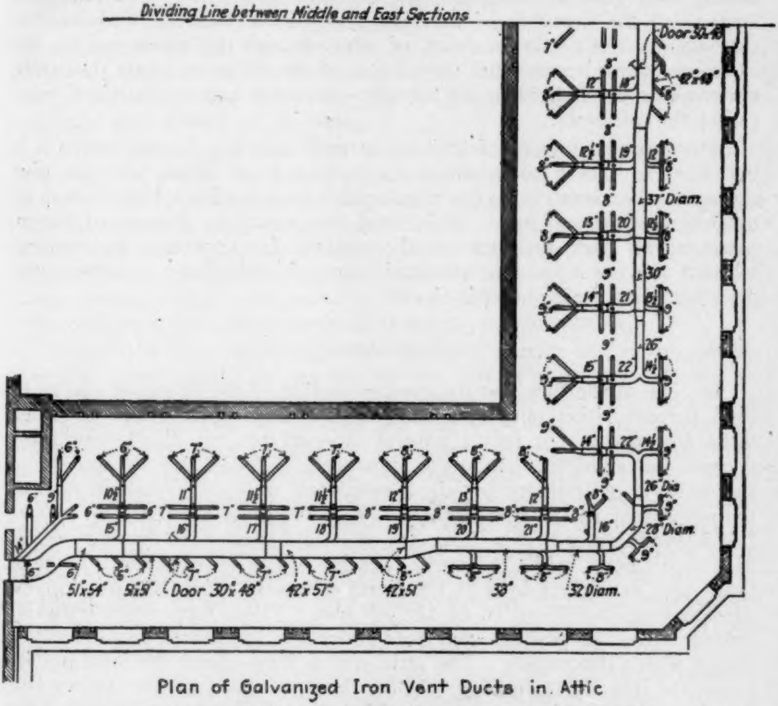


FIG. 6. TYPICAL ARRANGEMENT OF DUCTS IN ATTIC

and readily interpreted. Aside from this chart, readings are taken every two hours and kept on the daily log sheet of the plant, the records made being as follows: pressure and temperature of heater supply; pressure and temperature of heater discharge and outside temperature.

It will be noted that on all but the third floor, the radiators were placed in front of or under the windows, most of them being very long and low, extending practically the full width of the windows. It has been found that, while the amount of heating surface is no greater than required to heat the spaces with the existing window surface, it is proportionately no greater than on the third floor, the temperature is more readily controlled and is more nearly that which was originally intended. This is due primarily to the ideal location of the radiators and

the distribution of the heat throughout the full width of the window openings. This does not hold true on the third floor and it is here that considerable difficulty has been experienced in properly heating these rooms when the temperature of the water is sufficient to heat the other floors to the desired temperature. The third floor conditions are extremely poor because of the fact that in placing the radiators and determining their general dimensions the designer was seriously handicapped because of the demands of the occupants of the building who insisted that no space directly in front of the windows be taken up by the radiators. This necessitated the placing of the radiators along the walls, either along those dividing the adjacent rooms or against the piers separating the windows.

The care and upkeep of such a system is very low in cost where it is necessary to have a competent man, on duty at all times, who can give the necessary attention to the regulation and operation of the valves as outside conditions require. It is most economical in the use of steam, produces no back pressure on the engines and prevents overheating without the use of costly automatic control and allows a lower temperature when the building is closed.

VENTILATION

In view of the fact that the greater portion of the Museum was to be used for exhibition purposes, consisting of large open spaces in which there was nothing to retard natural air currents, no direct ventilating system was provided. In order, however, to assist the air movement and removal of foul and hot air, a ventilating unit was placed in each of the outer ends of the three wings. These fan units are made up of two full housed motor-driven fans, the east and west fans having wheels 6 ft. in diameter, while the north fans are 5½ ft. in diameter. Each of these units are driven by a 25 hp. direct connected motor and are capable of removing 70,000 cu. ft. of air per min. These fans remove the air directly from the skylight chambers above the wings and from the attic spaces above the ranges. The attic spaces throughout the building are connected directly to the floors below through chases in the outer walls, together with certain openings provided directly from the ceiling of the rooms on the third floor. The air movement, with the formation of a slight vacuum in the spaces mentioned, is through the openings from the floors below which, being entirely open, allows the removal of foul air from all portions of the building. A typical layout of ducts in attic is shown in Fig. 6 while the details of a fan unit can be seen in Fig. 7.

In the wings there is placed a false ceiling approximately 5 in. below the underside of the roof, made up of ¼ in. sackett board, suspended from the ceiling, terminating about 5 ft. from the eaves of the roof and connecting to the skylight spaces by means of an irregularly shaped galvanized iron bend which allows the passage of air underneath the roof girders which surround the skylight spaces. This arrangement permits not only the ventilation of the building, but produces a cooling effect, so that the spaces are habitable in hot weather as shown in Fig 8. The

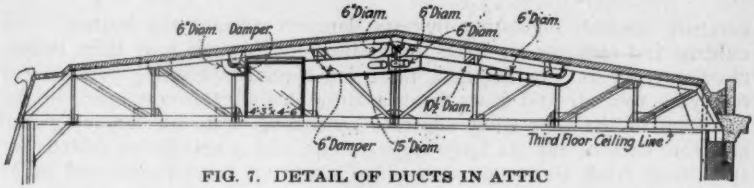


FIG. 7. DETAIL OF DUCTS IN ATTIC

treatment above the ranges performs the same duty, but is done in a different way, as the conditions are entirely different. These spaces are not used for any purpose except for ventilating ducts, cables, pipes, etc., and are kept closed at all times.

From the fan chamber there is run a brick duct across the width of the wing to the entrance of the attic spaces above the ranges. At this point a large galvanized iron duct, rectangular in shape at its beginning and later transforming to a circular duct to which are connected numerous branch pipes or ducts removing the air from all parts of the ranges. In each space of 18 ft. 6 in., there are 12 inlets so proportioned that each inlet removes the same quantity of air and so placed that the air is removed at points where pockets or warm currents collect.

All of the units mentioned above are used during the warm weather, but for ventilating purposes during the winter only one unit is used, namely, the one in the north wing, which, however, can remove foul air from all portions of the building, the movement in the ducts mentioned above being reversed when the natural flow of air is not sufficient to assist its passage through the fans which are idle.

Where special conditions existed, there were provided means for direct ventilation such as in the auditorium, which has a seating capacity of 565. In this case a 6 ft. full house fan, directly connected to a 15 hp. motor furnishes fresh air which is first washed by means of an air washer, then heated to a proper temperature by means of indirect cast-iron heaters. The air is then forced through ducts and flues to a large plenum chamber underneath the auditorium floor. To each chair leg an opening is provided through the floor, allowing the air to pass into the auditorium through perforated registers forming part of the chair legs. Provision is made for furnishing 30,000 cu. ft. fresh air per min., the temperature of which is regulated by means of an automatic tem-

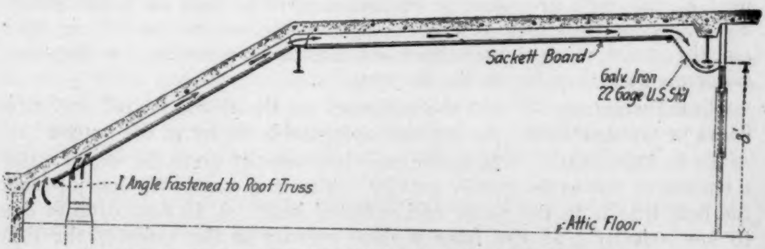


FIG. 8. FALSE CEILING USED AS VENTILATING DUCT

perature control, regulating by-pass dampers around the heaters. An exhaust fan removes air from the auditorium through four large ceiling circular registers. In addition, there are special ventilating systems for removing foul air and gases from a large alcoholic storage space in the basement of the west wing, separate ventilating units for removing foul air from each of the six large toilet rooms, and a ventilating system for furnishing fresh air to and removal of warm air from engine and boiler rooms.

DISCUSSION

C. R. DENMARK: The 2½ in. main in the small building in the East Court was put in afterwards and is connected with the supply that heats the air for the auditorium or tempers it and runs underneath the ground over to this building through a trench. From this main is a connection, 1¼ in. pipe, drop to each of the radiators; one goes in one end of the radiator and from the other end the pipe is connected back into the steam main and continues around the building making a complete circuit. The air is removed by automatic air valve and air trap. We have individual valves on the radiators but they are never used.

E. S. HALLETT: I was wondering how you regulated the temperature at the heaters. Did you put a thermostat outside or where?

C. R. DENMARK: There is no thermostatic control of the heating system of the building. The only thermostat is for the temperature of the auditorium. No means has been found to regulate the temperature of the water by outside temperature because there are too many elements entering into it.

T. H. IRELAND: I have found it worked out very nicely to take the Anderson or some other type of ball float trap and turn it upside down for an air trap.

C. R. DENMARK: The radiator connections were universal, top and bottom, and the flow of water is aided by the use of long turn fittings throughout the system. Where these connections are made if the drop is 1¼ in. the branch is 1½ in. at the radiator and it is a TY.

This T Y helps the flow of the water to pass through the radiator its entire length before the temperature drops at all. Of course sometimes the top is appreciably warmer than the bottom. The valves are never used on the radiator except in certain ones to be kept up permanently. The radiation decreases as you come from the top to the bottom floor and the radiators on the top floor are considerably smaller for the same exposure than they are at the bottom.

LEE PHILLIPS: Would the radiators on the third, second and first floors be warmer across the top and appreciably cooler at the bottom?

C. R. DENMARK: Yes, as the very low velocity gives the warm water a chance to rise to the top by gravity. Also it is due to the loss of heat, the heat that is in the water and radiated heat. A 10 deg. drop is due to low velocity. If you have a slow velocity in the radiator the first column would get hot long before the last.

No.648

HEATING AND VENTILATING THE BUREAU OF ENGRAVING AND PRINTING

By Nelson S. Thompson[1], Washington, D. C.

Member

THE present building of the Bureau of Engraving and Printing and its mechanical equipment were designed by the office of the Supervising Architect, Treasury Department, and erected under the supervision of that office in 1913. It houses one of the most interesting manufacturing plants in the country and is well worth a visit, not only on account of the architectural and mechanical equipment features of the building, but mainly to observe the manufacture of the paper money, postage stamps, etc., issued by the Federal Government.

It is not the province of this paper to describe the complicated processes in use in the manufacture of the output of this great plant, but merely to discuss briefly the heating and ventilating apparatus, which has proved satisfactory in service.

The building is comb-shaped, as shown in Fig. 1, with the head house approximately 500 ft. long by 68 ft. wide, and projecting from one side, four wings each of which is approximately 228 ft. long by 60 ft. wide, so that ample natural light and cross ventilation are assured. The total cubic content of the building is approximately 10,000,000 cu. ft.; the total glass area, set in metal window frames, is approximately 110,200 sq. ft., and the total wall area, exclusive of glass, is approximately 140,000 sq. ft., the wall having an average thickness of 18 in. and constructed of brick, faced generally with limestone as shown in Fig. 2.

There are installed for the purpose of heating the building, a total of 55,259 sq. ft. of direct cast-iron steam radiation which is connected on the two-pipe vacuum system, the layout of which is indicated in plan Fig. 1. There is also installed ventilating equipment with indirect heating apparatus for operation on the hot-blast system, but the installation is designed essentially for heating by direct radiation. It is the intent of the design to deliver all air for ventilation at 70 deg. fahr., for which the equivalent of direct radiation installed, is 66,000 sq. ft. approximately. There is, however, sufficient reserve indirect radiation installed

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Paper presented at the Annual Meeting of the American Society of Heating and Ventilating Engineers, Washington, D. C., January, 1923.

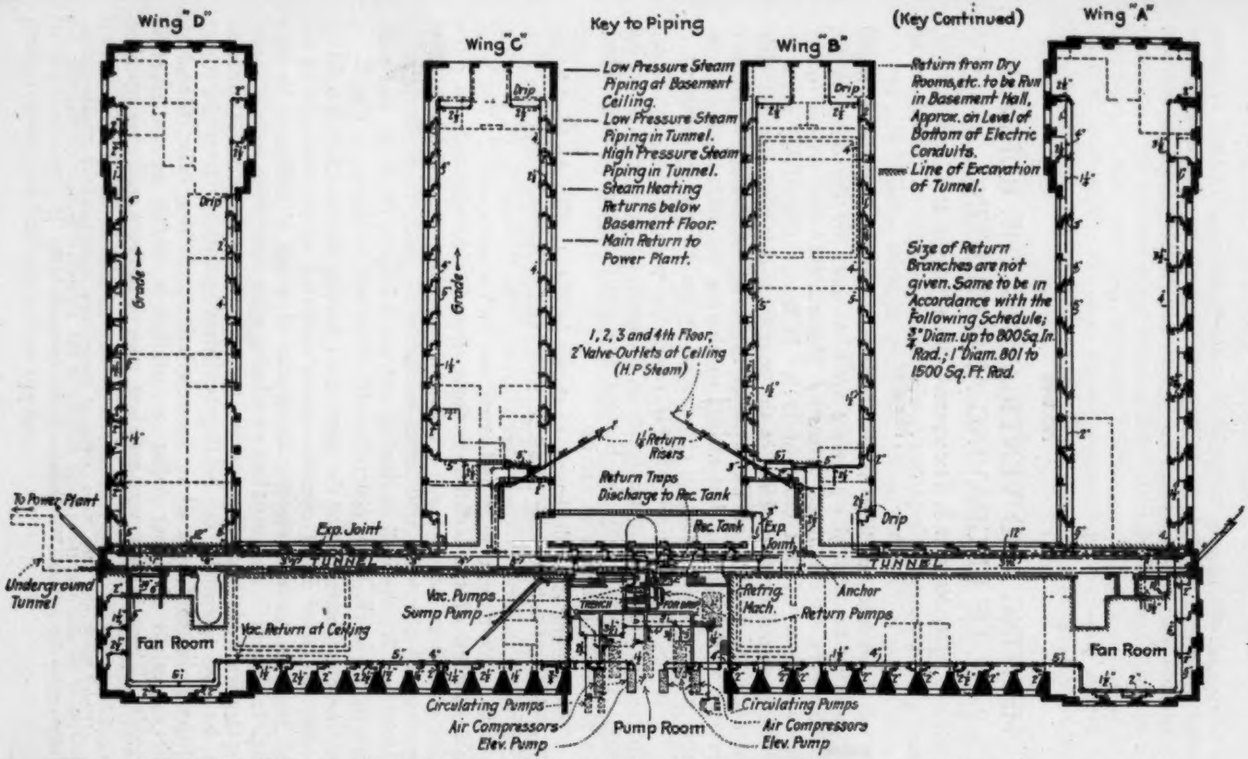

FIG. 1. BASEMENT PLAN OF BUREAU OF ENGRAVING AND PRINTING BUILDING, SHOWING PIPING LAYOUT AND ARRANGEMENT OF MECHANICAL EQUIPMENT

to permit of raising the temperature considerably higher than 70 deg., so that it may be used for heating the building in case of accident to the direct radiation system. The total equivalent direct radiation installed in the building is thus approximately 121,000 sq. ft.

Steam for heating the building and for the various industrial processes involving the use of steam, such as drying the paper, etc., is generated in a power house about 500 ft. distant and is conveyed by a system

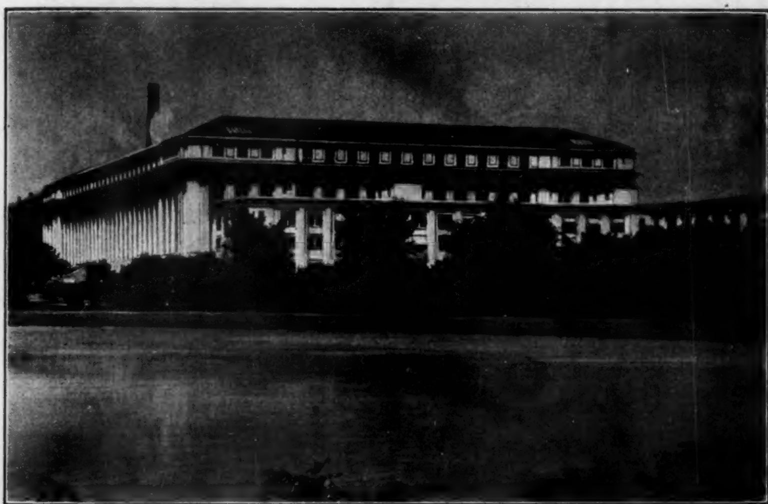

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FIG. 2. BUREAU OF ENGRAVING AND PRINTING, WASHINGTON, D. C.

of piping through a tunnel to the building, where the pressure is reduced to 5 lb. or less for heating and ventilating purposes.

There are many uses for steam in the building, one of the most important of which is the drying of paper as applied under the different conditions involved in the manufacture of paper money. Reference to this was made in a paper by H. C. Russell, Washington, D. C., entitled Drying Money, which was presented at the Semi-Annual Meeting of the Society in July, 1917, and appears in Vol. 23, 1917 of the TRANSACTIONS, page 571.

AIR SUPPLY SYSTEM

The ventilation system of the building, while very large in size, is extremely simple. It consists of two plenum fans, each 102 in. in diameter equipped with indirect heaters, air washers, and duplicate steam engine and electric motor drives. The steam engine drives are operated in the heating system so that the exhaust steam may be utilized in the indirect heaters; at other times in the year when heating is not required, the fans are driven by the electric motor. Each fan is equipped with

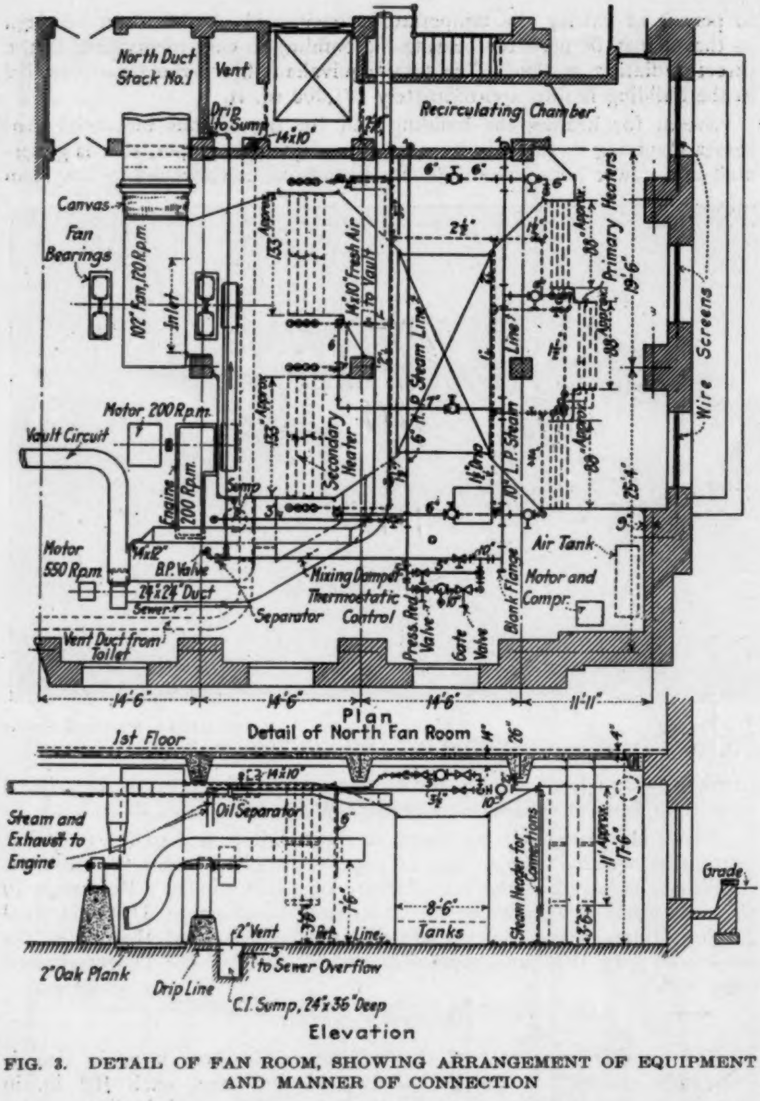

FIG. 3. DETAIL OF FAN ROOM, SHOWING ARRANGEMENT OF EQUIPMENT AND MANNER OF CONNECTION

an indirect heating chamber which contains approximately 11,000 sq. ft. of heating surface, which is under thermostatic control so as to deliver air at a temperature of from 75 to 80 deg. fahr. Details of a typical fan room are shown in Fig. 3.

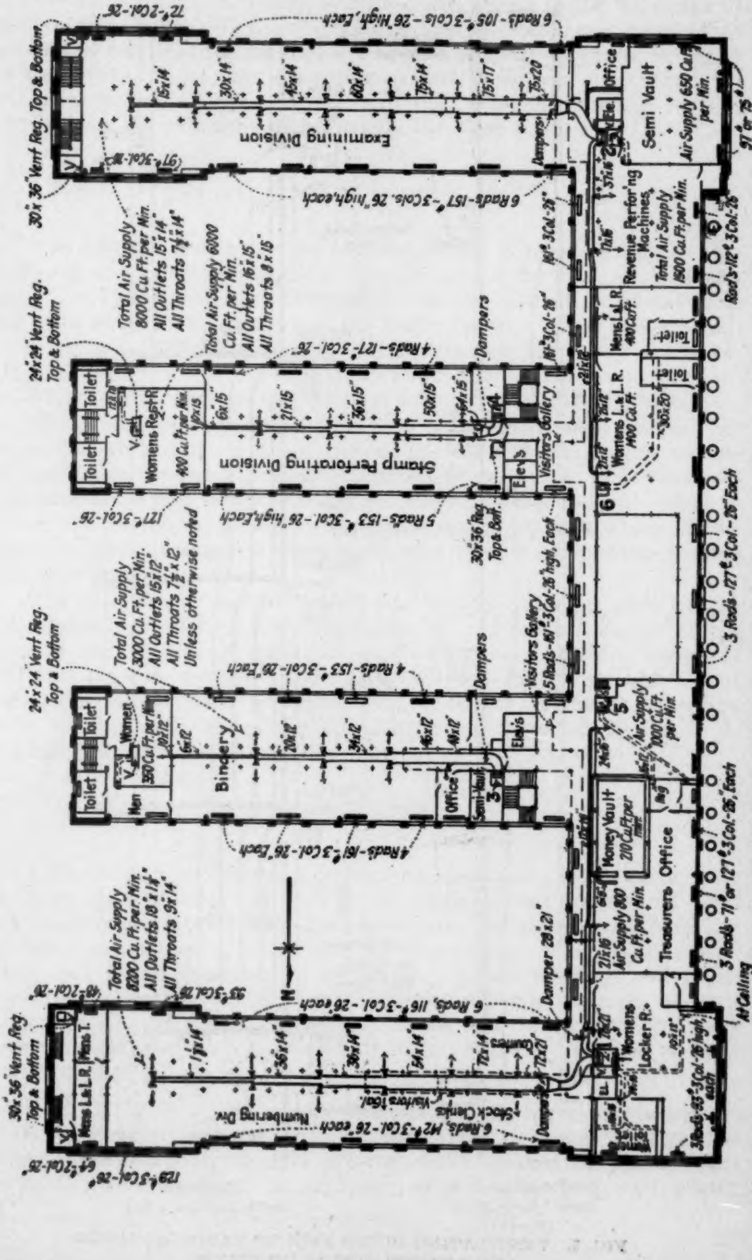

FIG. 4. SECOND FLOOR PLAN, SHOWING VENTILATION DISTRIBUTING SYSTEM

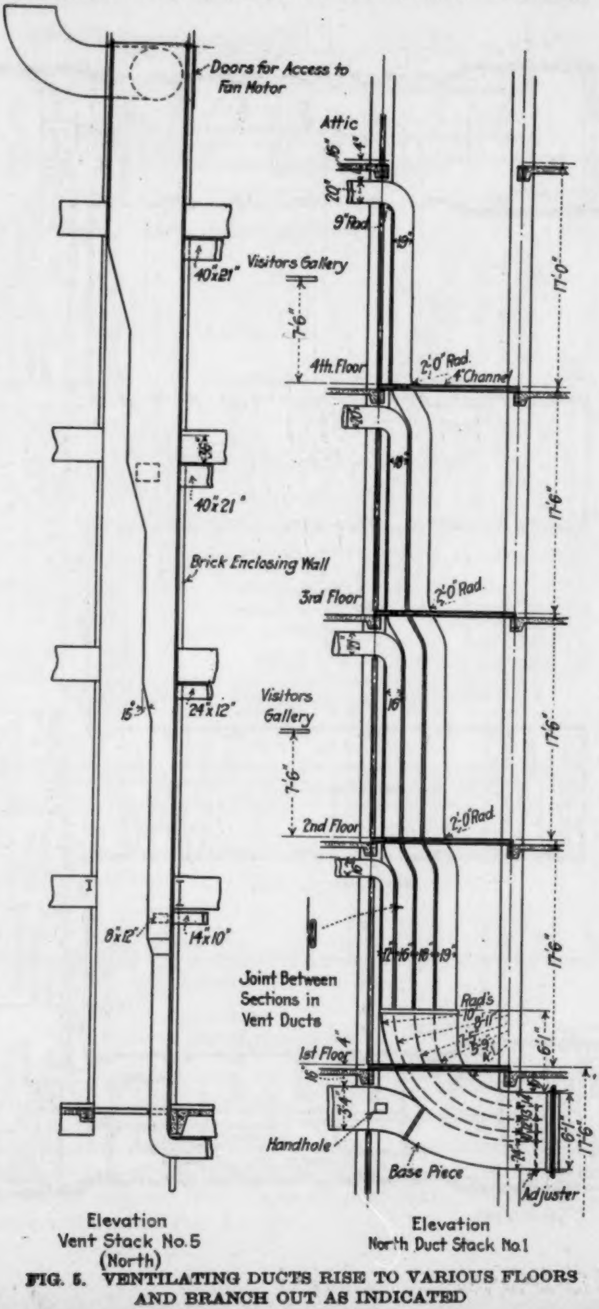

FIG. 5. VENTILATING DUCTS RISE TO VARIOUS FLOORS AND BRANCH OUT AS INDICATED

The ventilation distributing system is well shown in the floor plan of the building, Fig. 4. The fans discharge into vertical upcast ducts which are nested one duct for each floor. These ducts rise as shown in Fig. 5 to the various floors, one branching at each floor into the ceiling distributing system. The arrangement of the ceiling distributing ducts is shown with the side delivery outlets on the floor plans.

PLAN FOR EXHAUSTING AIR

The exhaust system for the building, which is operated through the locker rooms, is operated by disc fans located on the roof of the building. The system has been designed so that the exhaust air passes from the main workrooms out through the locker rooms. In the locker rooms there are grilles opening into the vent flue, which flue serves as the enclosure through which the vertical upcast duct from the ventilating fans in the basement rises to the delivery outlets to the various floors.

The numerous vaults in the building have ventilating apparatus entirely independent of the apparatus for the main building. The scheme of ventilation involved generally consists of forcing air into the vaults and permitting it to escape through the entrance doors into the rooms in which the vaults are located. The plate storage vault in which all the engraved plates are stored every night, has its own independent heating and ventilating apparatus.

The mechanical equipment for this building was installed by the Thompson-Starrett Co., New York, N. Y., and the installation of the heating and ventilating apparatus was under the immediate charge of William Driscoll, member of the Society. The duct work which is of interesting design, was all constructed from the engineer's drawings, built in Cincinnati, shipped to Washington, and erected by the Peck, Anderson & Peck Co., under the direction of Mr. Driscoll.

DISCUSSION

J. D. CASSELL: There is a determined effort being made, particularly in large office buildings of New York to show that the isolated plant was passing and that in the future most large buildings would have low pressure steam heating plants and all mechanical appliances, elevators, fans, etc., would be driven by electricity. Before the high coal prices we were producing electricity for nine-tenths of one per cent per kilowatt hour in conjunction with the heating season and the electrical company was then charging us $.04 a kilowatt hour for power and $.05 an hour for electricity.

E. S. HALLETT: We are all interested in the question whether we shall have motor drive or steam engine drive. It means a great deal whether we are going to have a plant in the building or whether it is to be omitted altogether. In St. Louis we still have cheap coal, paying

this year $4.19½ per ton. And yet if we buy electricity from the central station company to operate our fans the cost will be very close to the total cost of coal for all purposes in the building. There is no public service corporation that can compete with the isolated plant put up and operated as it ought to be, and using the exhaust steam for heating.

No. 649

SOME COMMENTS ON FURNACE INSTALLATION

BY F. T. GIBLIN[1], UTICA, N. Y. (Non-Member)
and
FRANK K. CHEW[2], NEW YORK, N. Y. (Member)

THIS paper has been prepared with the idea of adding to the valuable information on the proper design of furnace heating systems gathered by the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS. The various methods of determining heating pipe size in relation to the requirements of each particular installation, are compared in Table 1. The subject selected for consideration is the size and type of house in which the majority of American people live, which was erected under the hurried stress of war conditions, with every part of the heating system specially designed for the work to be done, in contra-distinction to those plants which under war conditions, had but one size of furnace, one size of pipe and one size of register to avoid complications and delays in installation regardless of the effect on the comfort, economy and permanency, so essential in a home heating system.

It will be noted that the proportion and relation of the parts compare favorably with the rules given in the paper presented before the Society by M. W. Ehrlich in 1918, with the proposed furnace code which has been under consideration during 1922, and also the Willard recommendation of large pipes to insure heating first floor rooms. The plans in Fig. 2 show a house of one of the eight or more types used in western Pennsylvania where one of the big steel companies had enormous demands made upon it and is responsible for a town of almost mushroom growth in 1917. One of the streets in the town which was located on a slope is shown in Figs. 1 and 3.

In these various houses, 800 Giblin standard all-cast-iron furnaces made by the Giblin & Co., Utica, N. Y., were used, and the piping system was laid out and installed by the heating engineers and contractors, the Mellot Furnace Co., Pittsburgh, Pa., as shown in the plans. The furnace has a 24 in. fire pot and a 45 in. casing, which has an area of 1590 sq. in. and a free area at the most restricted point of over 600

[1] President, Giblin & Company.
[2] Editor, *Sheet Metal Worker*.
Paper presented at the Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Washington, D. C., January, 1923.

TABLE 1. DIMENSIONS AND PROPORTIONS FOR COMPARING RULES

Room	Contents Cu. Ft.	Wall Surface	Glass Surface	Equivalent Glass Surface	Pipe Used Dia.	Pipe Used Area	Riser Size and Area	Proportion Pipe Area to E.G.S. Sq.in. Sq.ft.	Proportion Pipe Area to Space Sq. in. Cu.ft.	Proportion Riser Area to Space and E.G.S.	Heat Required 85 B.t.u. Sq. ft. E.G.S.	PIPE AREA: Ehrlich's Rule* Divide by 120 1st Fl. 150 2d Fl. Area Diam.	PIPE AREA: Willard's Rule* Divide by 100 1st Fl. 150 2d Fl. Area Diam.	PIPE AREA: Code Rule Applied
Upper and Lower Halls	1430	213	57	110		0					9650	78 (10) (78)	93 (11) (95)	1430 ÷ 800 = 1.8 213 ÷ 40 = 5 57 ÷ 12 = 5 12 × 8 = 96 12 × 9 = 108
Parlor	1701	208	72	124	12	113		1 to 1.1	1 to 15		10540	88 (11) (95)	105 (12) (113)	1701 ÷ 800 = 2.1 208 ÷ 40 = 5 72 ÷ 12 = 6 13 × 8 = 104 13 × 9 = 117
Dining Room	1701	288	72	144	12	113		1 to 1.27	1 to 15		12240	102 (12) (113)	122 (13) (132)	1701 ÷ 800 = 2.1 288 ÷ 40 = 7 72 ÷ 12 = 6 15 × 8 = 120 15 × 9 = 135
Kitchen	1192	180	54	90	10	78		1 to 1.27	1 to 15		8415	70 (10) (78)	84 (11) (95)	1192 ÷ 800 = 1.5 180 ÷ 40 = 4.2 54 ÷ 12 = 5 11 × 8 = 88 11 × 9 = 99
Hall Chamber	708	134	36	70	9	63	3½ × 12 = 42	1 to 1.1	1 to 11	1 to E.G.S. 1.7 Space 17	5950	39 (7) (40)	39 (7) (40)	708 ÷ 800 = 1 134 ÷ 40 = 3.3 36 ÷ 12 = 3 7 × 6 = 42
Parlor Chamber	1071	153	54	92	9	63	3½ × 12 = 42	1 to 1.46	1 to 17	1 to E.G.S. 2.2 Space 25	7822	52 (8) (50)	52 (8) (50)	1071 ÷ 800 = 1.3 153 ÷ 40 = 4 54 ÷ 12 = 5 10 × 6 = 60
Dining Room Chamber	1071	171	36	70	9	66	3½ × 12 = 42	1 to 1.25	1 to 17	1 to E.G.S. 1.9 Space 25	6715	44 (8) (50)	44 (8) (50)	1071 ÷ 800 = 1.3 171 ÷ 40 = 4.3 36 ÷ 12 = 3 8 × 6 = 48
Kitchen Chamber	756	135	36	70										
Bath Room	324	36	18	27	9	63	3½ × 12 = 42	1 to 1.52	1 to 17	1 to E.G.S. 2.3 Space 25	8405	56 (9) (63)	56 (9) (63)	1060 ÷ 800 = 1.3 171 ÷ 40 = 4.3 54 ÷ 12 = 5 10 × 6 = 60
TOTAL	9954	1518	465	815		556 2 Fl. 252	168	1 to 1.5 1st Fl. 1 to 1.6 2nd Fl.	1 to 18 1st Fl. 1 to 20 2nd Fl.	1 to E.G.S. 2 Space 25	69437	529 (567) 1st Fl. 191 (203) 2nd Fl.	595 (638) 1st Fl. 191 (203) 2nd Fl.	Total Factor 8 = 618 Total Factor 9 = 669 2nd Fl. 150

NOTE:—Return Air Supply 530 sq. in.
Grate 24 in. diam. = 452 sq. in. area or 3⅛ sq. ft.
Kitchen Chamber and Bath Room supplied by same pipe.
E. G. S. = equivalent glass surface.

*In columns giving results of Ehrlich's and Willard's rules, the first figures show the calculated results in required cross-sectionl area of pipe, and the figures to the right in parentheses, show respectively the actual pipe sizes selected and their corresponding cross-sectional areas.

sq. in. There are four 9 in., two 12 in., and one 10 in. warm-air pipes leading from the furnace, having a combined area of 556 sq. in. The air supplied to the furnace is taken from within the building and kept in circulation through a 30 x 30 in. register face and a 26 in. round galvanized return-air duct of 530 sq. in. area. The register box is so constructed as to facilitate the flow of air through the round pipe and is connected into a flaring duct, ending in a rectangular form reaching the width of the 45 in. diameter of the furnace casing. Floor registers are used in three down-stairs rooms of the sizes given on the basement plan. There is no warm air register in the hall, as the air returning from the different rooms to the return-air grill was considered sufficient to maintain a comfortable temperature in the hall, and this was, of course, a step toward economy of material and labor when both were in great demand.

FIG. 1. TYPES OF HOUSES HEATED BY WARM-AIR

Though 800 furnaces were used in eight or more different types of houses, more than general satisfaction has attended their service for several winters and no delays or annoyances have arisen from the use of different sizes of pipe, registers and furnace, as these details received the careful consideration that an engineer would naturally give. The amount of fuel consumed per season would be of interest, but the collection of such records was impracticable with no one at hand to take up the task. The factors that are known, however, are enough to warrant a study of the house, not only for its interesting equipment but more important for the suggestion of activities necessary for the better installation of furnace heating systems in the interest of those who occupy the homes.

The furnace, as indicated in Fig. 4, is of the all-cast-iron type with an outer galvanized casing and an inner casing or shield, having a space of about 1½ in. between it and the outer casing, and a greater space between it and the heating surfaces of the furnace varying as may be seen. The furnace was a 24 in. grate, with an area of 452 sq. in. or 3⅛ sq. ft. It is the custom of many manufacturers to rate furnaces on the basis of the warm air pipes area they will fill, and as a result the

home occupant risks having an unscrupulous builder use a furnace with a much larger casing than is placed on first class furnaces. Consequently, because of the much smaller grate and fire chamber provided in the casing of excessive free area, a larger area of warm air pipe cannot be effectively filled with air at the proper temperature without undue strain upon the interior parts or the furnace castings, nor will the fuel consumption be economical.

The basement plan shows the location of the furnace, the return air supply duct and the piping runs in the basement, while the other plans show the sizes of the risers and registers. It will be noted from second-floor plan that the wall stacks have numerous friction surfaces in the form of right angle turns which retard the air flow before reaching the

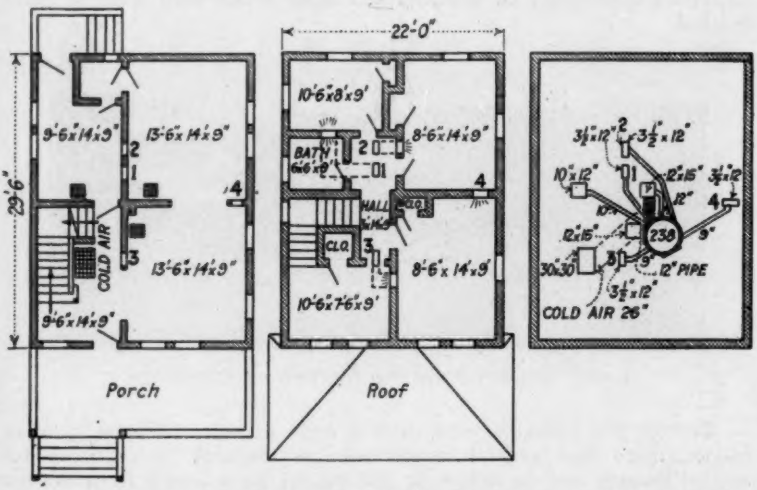

FIG. 2. FLOOR PLANS OF HOUSE SHOWING LAYOUT OF HEATING SYSTEM

register. This condition could have been avoided if those who drew up the house plans had designed the system with the thought of economical heating uppermost. The wall stacks are $3\frac{1}{2}$ x 12 in. in size and of better capacity and depth than are frequently found.

Table 1 gives the essential information on the cubic contents, wall surface, glass surface and the equivalent glass surface, counting 4 sq. ft. of wall as equal in cooling effect to 1 sq. ft. of glass, the size and area of the pipe and wall stack used, and the proportion to the space heated and glass exposure. The proportions are more generous than usual in the type of homes considered, which shows the influence of the steel company in question for better home comforts than the operation builder would have provided in such buildings. With the tabulation of contents and exposure, the comparison of the pipe area, riser sizes and capacity used can be made with those which would be determined by different

methods. This becomes a simple matter, and shows less discrepancy than might be expected all along the line.

At the Annual Meeting in 1918, an excellent paper entitled Engineering of Warm Air Furnace Heating, by M. William Ehrlich, was presented and gave for the first time to the Society, a comprehensive plan of determining the pipe area in warm-air furnace heating systems by a heat-unit method. A great deal of the information in Mr. Ehrlich's paper was based on information gathered from earlier articles of predecessors of the present *Sheet Metal Worker*, but the rule given was original and may be summarized as dividing the required heat units computed by any method, by 120 for first-floor rooms, by 150 for second-floor rooms and by 180 for third-floor rooms, at the same time making

FIG. 3. STREET IN PENNSYLVANIA TOWN SHOWING STYLES OF HOUSES HEATED BY FURNACES

an additional provision for rooms exposed to the northeast, the west and the northwest. With the equivalent glass surface, it was a simple matter to use the custom of the late A. H. Wolff of multiplying equivalent glass surface by 85, to obtain the number of B.t.u.s. that would be lost by 1 sq. ft. of glass surface with an in-door temperature of 70 deg. and an out-door temperature at zero. The late Prof. R. C. Carpenter of Cornell University, and Past President of the Society, popularized the custom of considering 4 sq. ft. of wall surface equal in cooling effect to 1 sq. ft. of glass.

The requirements for the different rooms in B.t.u. are given in one of the columns of the table and the provisions required for the first floor are given in another column as computed through Mr. Ehrlich's method, giving the area required, and the diameter and area of the pipe of the size nearest to that computed.

In correspondence with Professor Willard of the University of Illinois, Urbana, Ill., he has stated that he thinks the requirements for first-floor rooms should be determined by dividing the heat unit requirements by

100, while those for second and third floor rooms should be divided by 150 and 180 respectively. In another column are given the areas required under his rule or recommendation, and the diameter and area of the required pipe size.

Another column gives the method of determining the pipe area according to the code now under consideration by the Society. It will be noted that the cubic contents, the wall surface and the glass surface have been divided by different constants and the results added together and then multiplied by two different factors. The factor 8 was first decided upon, which required a temperature of 195 deg. in the air, although more favor is now being shown to the factor 9, which contemplates the temperature of 175 deg. at the register. These factors make comparison interesting.

The pipe area used on the system in the building, which has proved satisfactory, is 556 sq. in. By Mr. Ehrlich's rule 529 sq. in. would have been required, but in using the pipe sizes nearest and best adopted to the work, 567 sq. in. would have been required. Using the recommendations of Professor Willard 595 sq. in. would have been required, and using the pipe sizes nearest to those calculated, give an excess in area to 638 sq. in. Under the proposed code, using factor 8, 618 sq. in. of pipe area would have been required and using the factor 9, 669 sq. in.

The table also shows that there was provided an area of 252 sq. in. in the cellar pipe for supplying the second floor, while the riser area was but 168 sq. in. Both Mr. Ehrlich's and Professor Willard's methods of calculation for second-floor requirements show that 191 sq. in. of area were required making together with the required pipe area a total of 203 sq. in. Using the code rule 150 sq. in. of area would be required for the second floor. It will be noted that the total requirements of Mr. Ehrlich and Professor Willard agree for the second floor, but that there is a much larger provision made for the first floor by both Professor Willard and the code rule, than Mr. Ehrlich provides. This is probably due to the fact that the experiments at the University of Illinois which have influenced the code rule in some measure, have been the basis for Professor Willard's recommendations for much larger provision in the piping for first floors than thas been generally followed by the later day practitioners in the furnace field. It is more in harmony with the best work found in older heating systems.

It must also be noted that Mr. Ehrlich's rule the recommendations of Professor Willard and the code rule are based on taking the air supply from out of doors, while in the building under consideration the air supply is taken from within doors. At this point it is well to point out that in some sections of the country there is an opinion that when the air supply is taken from within doors smaller piping and smaller furnaces may be used. It is only necessary to point out that whether the air is taken from indoors or out of doors it does not change the cooling effect of a square foot of glass, and the pipe leading to the room where the exposure is made must be of a size that will bring air in sufficient quantity at the temperature required to offset the heat losses. Consequently there is little sound basis for reducing the size of the piping when the

inside air supply is used, even though it may be returned to the furnace at a 60 deg. temperature instead of at zero, if it is taken from out of doors. Neither is it desirable to reduce the size of the furnace, which would mean that it would have to be run harder and possibly more coal consumed.

It is far better to have the furnace of the size needed, as after all the size of the furnace is determined by the heat unit loss due to the exposure, and it must be of a sufficient size to burn enough coal to generate the heat units that will be required, based on a 70 per cent efficiency. Many

FIG. 4. VIEW OF INTERIOR OF FURNACE USED IN WAR-TIME INSTALLATION

furnace heating systems will show a much higher efficiency than 70 per cent. Heating systems in the operation built houses with small risers and small furnaces may succumb and need repairs more quickly when small furnaces and small friction producing wall stacks are used. These sometimes have less than half the area of the leaders from the furnace to them, to carry away the heat generated by the heating apparatus. In this reduction of the area of the riser as compared with the leader, may be found the reason for an expensive fuel consumption and the early destruction of the furnace.

It will be further noted in studying the table that there are 452 sq. in. of grate surface and 815 sq. ft. of equivalent glass surface, or a pro-

vision of 1 sq. in. of grate for 1.8 sq. ft. of equivalent glass surface. The grate surface is 3.14 sq. ft. and if 3 lb. of coal were burned per sq. ft. per hr. at an efficiency of only 60 per cent for anthracite coal, the heat required to offset the heat losses would be generated. To burn 3 lb. of coal per sq. ft. on a grate of 3 sq. ft. area is a simple matter with a chimney of good draft, and a very much larger consumption and a higher heat transmission can be readily procured if so desired.

Another way of determining the size of the furnace would be to assume that only 7200 B.t.u. would be derived from each pound of coal burned, and if 69,400 B.t.u. were required it would only be necessary to divide the 69,400 by 7200 to find that something over 9 lb. of coal per hr. would be required. Then to select a furnace with a grate surface that would burn 3 lb. of coal per sq. ft. per hr. for the work, regardless of the diameter of the casing, or depending upon it if the grate, fire pot and superstructure were of satisfactory size and arrangement, there would be little likelihood in the present day of finding a hide-bound furnace. A hide-bound furnace is one that is too restricted in the free area to permit enough air to fill the pipes and offset the heat losses that otherwise would equal the heat generated by the fuel consumed. Such furnaces are seldom found at the present time. Often the furnace with an excessive free area will in a measure do the work by becoming overheated, wasting a great deal of fuel, and finally have to be replaced by a furnace of proper capacity after the first one has been destroyed by overheating at the owner's expense.

It has been said that a warm-air furnace when driven becomes a hot-air furnace, but the furnace has four different capacities to meet ordinary weather, colder weather, zero weather and below zero weather conditions. They develop with the color of heating surfaces in a cast-iron furnace from the no-color capacity to the dull-red color capacity, canary color capacity, and the bright white capacity.

It is not expected that the plant in this particular type of building could pass without severe criticism, nor is it expected that the deductions and comparisons herein made are all that is to be said on the subject, but rather, that the paper as presented shall bring out a discussion that will either corroborate its findings or refute them. The engineering society is different from a commercial body, and it is through it that the search for absolute truth is expected.

DISCUSSION

J. D. HOFFMAN (written): All leader, stack and register sizes obtained from all rules are based on circulating air temperatures and have nothing to do with outside air temperatures, excepting as the outside temperature, say zero, is used in determining the original heat loss from the room. Concerning the furnace size, however, the question of inside versus outside air applies with force. A furnace selected with capacity just sufficient to supply the estimated heat loss from the house, would be too small to heat, in addition, the air supply from zero to say 65 deg.

fahr. (the temperature of the return air from the rooms to the furnace). Probably the safest solution of the furnace problem is to add to the estimated heat loss to the house 12,000 to 15,000 B.t.u. when obtaining the grate size. This allows for a possible admission of 10,000 to 12,000 cu. ft. of outside air per hour through especially prepared ventilating ducts if desired. In any case this is a safety factor which can do no harm and in many instances it may be badly needed.

The wall stack deserves serious consideration along with the pipe sizes and furnace grate. Here is where most of the furnace systems go wrong, and yet we keep on installing the same misfits year after year without any attempt toward remedying the evil. I suggest that all partition walls or sections of these walls, carrying heat stacks to second floor rooms, should have not less than 6 in. studding and that the least internal dimension of the included stack should not be less than 4½ in.

All theoretical design must be tempered with good judgment, and in no part of the engineering field is this shown so strikingly as in furnace heating design. Theoretical and experimental deductions are of the greatest value but these are necessarily modified by surrounding conditions. We have been trying so long to design and install furnace systems that figure out correctly for the fair-to-better type of house that it is time we began to think of the house with only fair-to-poor construction and the comfort of the people who have to live in them. When one stops to consider that a poorly constructed frame wall may easily lose 3 or more times the amount of heat that will be lost through a wall of good construction, and that on a zero day with a 15-mile wind blowing, three standard sized windows may give a sufficient amount of inleakage to cost 15 cents per 24-hour day to heat the air from zero to 70 deg., and further, that other faulty constructions cause additional heat losses, it is then and not until then that the full problem of furnace heating presents itself.

M. W. EHRLICH (written): This constructive presentation, which gives a typical example of 800 furnace installations, is certainly timely and brings to light some features of design, or furnace systems proportions, that are now before the Society in the form of the proposed code. The comparisons made between the actual installation and various rules and proposals give a basis of comparison that is easy to follow and should therefore merit close study and full discussion by designing engineers as well as furnacemen.

In analyzing this concise treatment it is found to be replete with information and lends itself to further dissection. On this premise, I have made a recapitulation of the physical conditions and some of the computed values, and present them here in tabulated form, with some modifications. My method of proportioning leader sizes permits the use of odd dimensions to the ½ in., and the figures are given accordingly under the heading of "Ehrlich's Rule."

In one case under Professor Willard's Recommendations the size has been changed from 8 to 9 in. to take care of the requirements. Then two sets of figures are given separately as those based on the proposed code,

and those based on the substitute article by Professor Hoffman. In each case the areas computed by the respective rule or method is listed, and also the nearest pipe size and its area, according to that method, all being

RECAPITULATION OF PIPE AREAS COMPARING INSTALLATION WITH RULES

Upper and Lower	Installation Pipe Used Dia.	Installation Pipe Used Area	Ehrlich's Rule Computed Area	Ehrlich's Rule Required Pipe Dia.	Ehrlich's Rule Required Pipe Area.	Willard's Computed Area Sq. In.	Recommendation Required Pipe Dia.	Recommendation Required Pipe Area	Proposed Code Computed Area	Proposed Code Pipe Required Dia.	Proposed Code Pipe Required Area	Hoffman's Computed Area	Substitute Required Pipe Dia.	Substitute Required Pipe Area
halls	0	0	78	10	78	93	11	95	96	12	113	108	12	113
Parlor	12	113	88	11	95	105	12	113	104	12	113	117	13	132
Dining Room..	12	113	102	12	113	122	13	132	120	13	132	135	14	154
Kitchen	10	78	70	9½	70	84	11	95	88	11	95	99	12	113
Hall Chamber..	9	63	52	7	40	39	7	40	42	8	50	42	8	50
Parlor Chamber	9	63	39	8½	50	52	9	63	60	9	63	60	9	63
Dining Room Chamber ...	9	63	44	7½	44	44	8	50	48	8	50	48	8	50
Kitchen, chamber and bathroom	9	63	56	8½	56	56	9	63	60	9	63	60	9	63
Totals		556	529		552			651			679			737
Furnace grate Area sq in..	24	452	529	26	530	?			?			340	22	380
1 □" leader to 1 □' egs....		1.46			1.47			1.25			1.2			1.1
1 □" leader to cu. ft.......		17.9			18			15.2			14.6			13.5
1 □" leader to B. t. u.....		124			125			106			102			94
1 □" leader to grate area □'		0.81	1.0		0.96			?			?			0.52

based on the figures in the original tabulation as presented in the paper under discussion. Just one feature, and that is grate area of furnace required, is not given in the paper for Professor Willard nor for the code.

The practical extremes that show up by this further analysis are as follows:

Warm-air pipe area, 522 sq. in. minimum and 737 sq. in. maximum.
Furnace grate area, 380 sq. in. minimum and 530 sq. in. maximum.
Ratio leader area to E. G. S., 1.1 sq. ft. minimum and 1.47 sq. ft. maximum.
Ratio leader area to contents, 13.5 cu. ft. minimum and 18.0 cu. ft. maximum.
Ratio leader area to B. t. u., 94 B. t. u. minimum and 125 B. t. u. maximum.
Ratio leader area to grate, 0.52 sq. in. minimum and 0.96 sq. in. maximum.

These figures certainly cannot be considered in close agreement, but undoubtedly there is a happy medium that would serve. Or still better, for all these figures a close practical result should be obtainable and surely the engineers here assembled are the ones best fitted to make a final decision.

This occasion is taken to thank the authors for their mention of the Ehrlich rules in this discussion. This method, presented in 1918, was original in that it gave for the first time the B. t. u. factors 120-150-180 as a simple basis of figuring pipe sizes for furnace systems. Whether or not the numerical values of these factors remain or are modified, the real point is to retain the method and change the factors to the right value, if they are incorrect, and to give credit where it is due.

A. C. WILLARD (written): In my opinion, a plant designed on a heat delivery basis with the factors of 100 B.t.u. for first, 150 B.t.u. for

second, and 180 B.t.u. for third-floor registers per sq. in. of leader, with no stack to second or third floor less than 75 per cent of the area of the connected leader area, and with a short return air duct, equal to at least 100 per cent of the leader area, will give complete satisfaction in the coldest weather with a register temperature of 175 to 180 deg. fahr. It is also true that leader sizes may be successfully reduced if you are willing to accept a register temperature of 195 to 200 deg. fahr., and increase the combustion rate. For such a condition the factors become 130, 195 and 240. The furnace efficiency will probably drop from about 60 to about 55 per cent at this higher register temperature.

We have never at any time recommended or suggested that risers or stacks should have 75 per cent of the capacity of the connected leader. Our tests showed that if the stack area was 75 per cent of the leader area (using a 10 in. leader and a 4⅜ x 13 in. stack), then the air passing capacity or heat delivery capacity of the system for a given register temperature was a little over 90 per cent of the capacity of the same 10 in. leader connected to a 10 in. diameter stack. Such a combination does not seem to us at all undesirable in practical work, provided the first-floor leaders are of ample capacity, and that means such leaders must be larger than are usually installed today.

F. R. STILL: Professor Willard showed in Cleveland by curves that the size of leaders is dependent entirely upon the temperature; in this paper the temperature at the register is entirely omitted. The two are not very far apart, however. We now have the data and it is very easy to get the temperature.

FRANK K. CHEW: So far as this paper is concerned, I doubt if anybody ever measured the temperature at registers in any of the 800 houses mentioned so that that information is not available.

In so far as Mr. Ehrlich mentioned temperatures in his discussion, the temperatures at the registers that he contemplates, I know, are very similar to those Professor Willard contemplates.

P. J. DOUGHERTY: This so-called Ehrlich rule advocated by the last speaker is a typical example of the hit-or-miss rules, mostly miss, that clutter up furnace literature. As explained in the October, 1916, *Metal Worker*, this 120, 150 and 180 leader area rule is based on a temperature of 130 deg. at the register and an air velocity in feet per minute in the leader for first floor of 262, second floor 327 and third floor 392. The actual facts as demonstrated at the University of Illinois, with a register temperature of 130 deg., the heat units per sq. in. of leader area are to first floor 43, second floor 85 and third floor 105, which is far from first floor 120, second floor 150 and third floor 180 as claimed by Mr. Ehrlich. At 140 deg. register temperature as demonstrated at Urbana the velocity in feet per minute to first floor is 120, second floor 220 and third floor 270, which is far from first floor 262, second floor 327 and third floor 392 at a much lower temperature.

This is a fair sample of the worthless rules given the furnace trade as the engineering of warm air furnace heating. The warm air

furnace manufacturers of the country deserve a great deal of credit for the extensive research work on warm air furnaces that they have been carrying on for several years at the University of Illinois. As a result of this work, furnace heating is being placed on as sound an engineering basis as boiler heating. There is no excuse any more for such wild guessing as illustrated in the so-called Ehrlich rule. The 9, 6, 5 leader rule is based on real engineering data and when used in conjunction with the heat loss rule of 40, 12 and 800 (1922 GUIDE, p. 97), the figuring of a furnace job is reduced to an engineering basis that can be easily grasped by the average furnace dealer.

R. A. SMALL: We have a peculiar problem in Wisconsin. We are now preparing a code in which it seems necessary not only to require certain services in a house but certain performance to get that service. Manifestly it is only part of our job to require a certain room temperature and a certain degree of ventilation in a schoolhouse, unless we also require a performance that will give us that service. To require 100,000 B.t.u. in a room is no good with only 40,000 B.t.u. capacity in your furnace. Therefore we are developing a sort of formula from which we can determine the furnace capacity. The designer is expected to go to a building or examine his plans and study the job so that he knows how much service has to be put in that building and then select an outfit that will produce that service.

First, we assume adequate chimney size and height, 12,000 B.t.u. soft coal for fuel, 60 per cent over-all efficiency, and the old reliable 30 square feet of heating surface for each square foot of grate, with base combustion rate of 8 pounds per hour for round pot furnaces and 10 pounds for horizontals. We allow 1 per cent bonus or demerit for each unit of ratio above or below 30 to 1 up to 40 to 1 and down to 20 to 1; so that a round pot surface having 4 square feet of grate and 80 square feet of heating surface would be rated at 4 x 8 x 90 x 60 x 12,000=about 207,400 B.t.u. per hour and could be used on that much or less load. If the heating surface were increased to 140 square feet, the rating would be 4 x 8 x 1.05 x 60 x 12,000=about 241,900 B.t.u.

In that way we attempt to difference between a furnace that simply burns coal and one that puts the heat of combustion into the air of the heating system. The question is, are we on the right track and are we going to get the results desired?

J. A. DONNELLY: How much could you vary the length of firing period and still get the factor you use?

P. J. DOUGHERTY: That depends upon the size of building or residence—8 to 12 hours are the practical limits.

H. L. WILLIAMS: We take the 30 to 1 ratio not as a flat rate but as some place to start from. You can't go to work and wholly decide your rate of combustion because that is an essential feature in burning coal. Therefore we are trying to get down to some sort of principle, right at the very point where our rate of combustion is also related to our stack draft, and prepare a chart.

No. 650

HEAT TRANSMISSION THROUGH BUILDING STRUCTURES

By E. F. Mueller[1], Washington, D. C.

Member

THE status of this work is such that it is impossible to give a report of results so just an outline descriptive of the method used will be presented.

The general problem involved in an investigation of heat transmission through walls is to determine the heat flow through a given wall, when the air on one side of the wall is at a temperature, t_1, and the air on the other side is at a lower temperature, t_2. Under such conditions a survey of temperatures would disclose that the temperature falls continuously from t_1 *to* t_2, with abrupt changes in the rate of change at the surfaces, but without discontinuities in the temperature itself. There will be, in general a considerable difference between t_1 and the temperature at the warmer surface of the wall, and a corresponding difference between the temperature of the colder surface and t_2, the remainder of the temperature drop occurring in the wall itself.

TWO METHODS

There are two methods by which the problem may be attacked. In the first, the effects of the several resistances which retard the flow of heat are determined separately. In the case cited, it would be necessary to determine, for example, what difference between the temperature t_1 and that of the warmer surface would be necessary to cause unit rate of heat flow per unit area of surface. The same determination must be made for the other surface and the air in contact with it. Finally, the temperature difference which must be maintained between the two faces of the wall in order to produce unit rate of heat flow per unit area, would have to be determined. These temperature differences are proportional to the resistance to heat flow. From complete data of this kind it would

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be possible to compute the heat flow through any wall for any given set of conditions. This method may be called the analytical method. Taking into consideration the large number of combinations which can be made with even a few elements, this method would appear to be the one likely to prove most generally useful, and most of the heat transmission work done at the Bureau of Standards has been directed toward obtaining data for the use of this method, and this work will be continued.

In some cases the analytical method described is not easily applied, and, in any case, the results obtained will command more confidence if they are verified by the second method, which involves a direct determination of the heat flow and temperature differences for a complete structure, such as a brick or hollow tile wall with plaster, etc., as usually built. It is the purpose of this paper to describe how this second method is being applied. Obviously the results obtained in this method will depend not only upon the properties of the structure under test, but also upon conditions as to air circulation.

THEORY OF SECOND METHOD

The method is most simply explained by reference to fundamental ideas of heat conduction. Assume that a constant temperature difference is maintained between the two faces of a flat slab of material of uniform composition, then, after a steady state has been established, the quantity of heat, dq, which will flow across a given area, A, in a given time interval, dt, is proportional to the area, to the time interval, and to the temperature gradient, or temperature drop per unit distance, in the direction of heat flow. If the direction of heat flow be taken as the axis of x and temperature be denoted by T, the temperature gradient is dT/dx. The constant of proportionality which makes it possible to convert the statements in reference to proportionality into an equation is the thermal conductivity c, of the material. The equation for heat flow may therefore be written:

$$dq = c\,A\,\frac{dT}{dx}\,dt$$

$$\text{or}\ \frac{dq}{dt} = c\,A\,\frac{dT}{dx}$$

For a number of years, measurements of thermal conductivity have been under way at the Bureau. In these measurements, the rate of heat flow, dq/dt, the area, A, of the specimen, and the temperature gradient, dT/dx, were measured, thus determining the conductivity, c, by means of the above equation. It is obvious that, if c has once been determined, it is possible by measuring dT/dx, to determine $\frac{dq}{dt}/A$ or the heat flow per unit area. It is thus possible to make an instrument for measuring heat flow, and such an instrument has been constructed. It consists of

a piece of wood eight inches square, and about 5/8 in. thick, with a large number of thermojunctions on each face. These thermojunctions are connected in series so as to indicate the difference between the temperatures of the two faces. These faces are covered with a thin veneer of wood to protect the junctions. This instrument, which was constructed about five years ago, has been named a conductimeter. The conductimeter was calibrated by using it as a specimen in the thermal conductivity apparatus. After this calibration it is possible to place this conductimeter against the wall of a room, and since the heat which flows through the conductimeter must also flow through the wall, to measure the heat flow through the wall, under the existing conditions, modified, of course, by the presence of the conductimeter. From such measurement it is possible, in principle, to deduce the transmission data for the wall. An attempt to apply the instrument in this way would at once reveal that the interpretation of the data is greatly complicated by the fact that steady conditions do not obtain. The wall, in addition to transferring heat, acts also as a reservoir to alternately store up and discharge heat.

It appeared desirable, as a first step, to investigate the performance of the conductimeter under favorable conditions. Instead of using it directly on walls of buildings already constructed, the experiment was brought into the laboratory, where conditions could be controlled.

CONDITIONS OF LABORATORY TEST

Wall specimens, 32 in. wide and 6 ft. high, were built. These specimens are used as the partition between a warm-box and a refrigerated-box. These boxes are insulated with cork, and have means for circulating the air to provide uniformity of temperature. The edges of the specimen are also insulated. The warm side of the specimen is covered with a wooden panel, of the same thickness as the conductimeter, and having an opening at its center to receive the conductimeter. The temperature in each box is automatically controlled.

The indications of the conductimeter serve to determine the rate of heat flow, and the temperatures, both in the air and at the wall surfaces, as well as inside the wall, are indicated by thermocouples.

Measurements are to be made on most of the ordinary types of wall construction, such as frame walls, brick walls, several types of concrete walls, hollow brick walls, etc.

It should be emphasized that in work of this kind the quantity to be measured is not a physical constant or even a constant of the structure which is subjected to measurement. If the result of such a measurement were to be given as a single transmission constant, the value given would have to be hedged about with a large number of restrictions as to conditions of air flow, exposure, etc. The complete investigation will therefore involve a determination of the magnitude of the variation of the transmission through a given wall, under varying conditions.

The very fact that the results of work of this kind are so dependent upon the conditions under which the tests are made is one of the very

good reasons why such work should be done, not in one place or by one organization, but at a number of places. A few years ago, the idea of organization in research, in order to avoid duplication, became very popular. There has been a necessary and justifiable reaction against the extreme to which this doctrine was urged, because work which is not duplicated must always be subject to doubt as to whether it can be duplicated. This is particularly true in work of the kind under discussion, where the results may be profoundly modified by apparently trivial changes in the conditions under which measurements are made. The results obtained will command confidence only to the extent that similar or comparable results can be obtained in other laboratories under different conditions. It is our hope to contribute our proper share toward the solution of the problems of heat transmission.

DISCUSSION

PERCY NICHOLLS: The chief point of the heat transmission measuring meter or conductimeter is what accuracy and what unchangeableness in it can one guarantee. Personally I believe there is a very large opportunity for this type of plate, not particularly for test laboratory use but rather for testing outside, and, having one of these plates it will be possible to get the lag in the flow of heat under natural conditions. I think also that it will do a great deal toward restoring the confidence of engineers in heat transmission values.

L. A. HARDING: The variation of density in building materials is such that two experimenters in different localities will ordinarily not check. This is not due to the fact that the tests are not accurately run but rather to the fact that the variation in density of the material in question is such that we can never expect to get check results. For instance, you take brick from the same kiln at the same time and they will run a soft brick and a hard brick. Those tests will vary from 5 to 8 per cent. The same is true of lumber. You find the same difference in concrete. We have run tests on 1:2:4 concrete and found a variation of 10 per cent from the results obtained at some other point.

W. H. CARRIER: One point that should be given consideration is the heat lag in the walls that is caused by variation of temperature. Especially in cold storage where we have very thick walls, there may be a lag. A steady temperature may not be secured for 24 hours and by that time the temperature has changed and you have got something else. The walls also being cold at certain times give a great deal higher transmission than that calculated. There is a point where this method, of course, is of value as it shows the true heat absorbing effect of the walls which require a given amount of heating or refrigeration. At the same time it seems that this as a meter, unless we have some method of recording density, would hardly be practicable.

H. M. HART: We are not only interested in the wind velocity but we also lack tests to tell us the effect of the wind on the various surfaces and the difference in effect of a direct wind and a side wind.

MR HEINZ: The practical engineer wants the simplest thing he can possibly get. How accurate would a device of this same general character be? Is this method of heat loss reasonably accurate for practical use?

E. F. MUELLER: I think as has been emphasized by several speakers this morning that the difficulty is not in the measurement but in specifying what is being measured and as long as you have that difficulty it is never going to be possible to get extremely precise results. The measuring instruments that we have are easily capable of the required precision, I believe. Without having tried I should say that if the measuring instrument were properly designed the precision of it would possibly be higher than it was possible to determine the condition under which the heat loss occurs. That is, the variations due to varying wind velocities and exposures, absence or presence of sunshine and things of that kind would be factors, which would make the results obtained at different times differ from each other, rather than the lack of precision in the measuring apparatus itself.

W. H. CARRIER: From a practical engineering standpoint we want accuracy for the determination of heat losses through walls. The heating and ventilating man wants to know what his theoretical losses through the walls are alone. We have a big loss through infiltration. If we figure on the heat losses alone, we would be away off on a good many of our buildings. The factor of variations due to air circulation and surface factors are of great importance, especially with thin walls, glass and corrugated steel buildings. If the engineer could have placed at his disposal a large number of selected tests on different types of buildings, analyzing the losses into their component parts—so much for walls, so much for infiltration—under different conditions so that he would be able to say exactly what was going on and get a complete history and diagram of the way those heat losses occur, it would be a big step forward and more important than the calibration of losses through the walls themselves.

No. 651

THE TESTING OF ANEMOMETERS

By. O. J. Hodge[1], Washington, D. C.

Non-Member

ANEMOMETERS are used to a large extent in connection with ventilation measurements in mines and buildings. In these measurements the anemometer is used for determining the velocity at different points of the cross-section of the shaft through which the air is flowing. In making the measurement the instrument is held in the stream of air and the difference between the initial and final readings of the dial is noted over a period of time which is measured with a stop-watch. To facilitate the reading of the anemometer at the start and finish of the run, the instrument is equipped with a means for engaging or disengaging the indicating mechanism from the rotating vane-shaft.

In the calibration of the instruments at the Bureau of Standards the process is reversed, when the velocity does not exceed 2500 ft. per min. The anemometer is moved while the air is stationary. This is accomplished by means of a whirling arm.

In general, the anemometer is mounted on a thin metal plate projecting from the end of the arm, with the axis of the instrument tangent to the circle of travel. The arm is then rotated at a constant speed for a fixed length of time and the number of feet registered by the anemometer noted. The true speed of the arm at the circumference of the path through which the axis of the anemometer travels having been measured, the data thus obtained are used in determining one point on a curve, Fig. 1, in which the ratio of the true air-speed to the speed recorded by the anemometer is plotted as a function of the anemometer reading. Therefore the true air-speed is obtained by multiplying the speed determined by the anemometer by the correction factor as read from the curve. If this curve were horizontal throughout its entire length it would of course be sufficient to calibrate each anemometer at one speed only, but as the curve is far from horizontal, particularly for the lower speeds, it is necessary to calibrate the instrument over its entire range. It is evident from

[1] U. S. Bureau of Standards, Washington, D. C.

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Paper presented at the Annual Meeting of the American Society of Heating and Ventilating Engineers, Washington, D. C., January, 1922.

the shape of the curve that the maximum degree of accuracy to which the anemometer may be used occurs at the higher speeds where the change in the correction factor is comparatively small for change in anemometer speed. As the speed decreases the curve becomes steeper and at a speed in the neighborhood of 200 ft. per min. the curve, for anemometers of this type, is so steep that the relation between the anemometer reading and the true speed cannot be determined with any degree of accuracy. Accordingly, the lowest speed at which the anemometer should be used is about 200 ft. per min.

The curve shown in the chart is for a typical low speed anemometer designed for the measurement of air speeds not greater than 2000 ft. per min. In the case of instruments designed for the measurement of higher speeds, the increased range is obtained at the cost of decrease in sensitivity so that the uncertain region shown by the steep part of the curve occurs at considerably higher speeds than 200 ft. per min.

In detail, the apparatus for making the calibration is as follows: The power for rotating the arm is derived from a motor which is operated on a 120 volt storage battery line in order to insure uniform speed. The motor is geared to the top pulley of a pair of cone pulleys which are connected by a belt. From the left of the lower cone pulley the drive is transmitted by a sprocket and chain to a sprocket mounted on a horizontal shaft.

The horizontal shaft drives a vertical shaft by means of a worm and gear such that the vertical shaft with the arm revolves only one-sixteenth as fast as the horizontal shaft. The radius of the circular path travelled by the anemometer is about 10 ft. This radius and the reduction between the worm and gear at the base of the vertical shaft being known, the speed of the anemometer with reference to the ground is obtained by measuring the rate of rotation of the horizontal shaft. The number of revolutions of this shaft is measured with a revolution counter operated from an eccentric on the shaft. The time is measured by minute signals obtained through the Bureau's master clock circuit.

Variation of the speed is accomplished first by shifting the cone-pulley belt by a hand crank. By this means a gradual variation of the rate of rotation of the lower cone pulley may be obtained, varying from a rate half as fast to a rate twice as fast as that of the upper cone pulley. A wider range in speed is obtained by changing the chain drive to the horizontal shaft. There are two chains with two sets of sprockets such that with one chain the rate of rotation of the horizontal shaft is half as fast as that of the lower cone pulley, and with the other, twice as fast. To change from one chain drive to the other a pin is removed from one of the sprockets on the horizontal shaft and inserted in the other sprocket so that the latter drives the shaft while the first merely turns loosely around the shaft. To obtain the lowest speeds, that of the motor is reduced by a rheostat and for extremely small variation of speed a smaller variable resistance is used. The apparatus is set for a desired speed by means of a liquid tachometer connected to the horizontal shaft by a spring belt. Although the scale of the tachometer is calibrated to give readings cor-

responding to the speed in feet per minute at the end of the arm where the anemometer is fastened, this measurement of the speed is only approximate. The actual speed is determined, as mentioned, by means of the revolution counter and master clock circuit signals.

By the methods described above, a range of constant speeds at the end of the arm is obtained varying from a value less than 50 ft. to a maximum of 2500 ft. per min. The lowest air-speed at which the vane wheel of the usual type of anemometer will revolve continuously is rarely less than about 75 ft. per min. Calibrations at higher speeds than 2500 ft. per min. are made in the Bureau wind tunnel in which instruments have been tested for speeds up to 10,000 ft. a min.

In making a run with the whirling arm apparatus, the anemometer and revolution counter readings are recorded and the arm then made

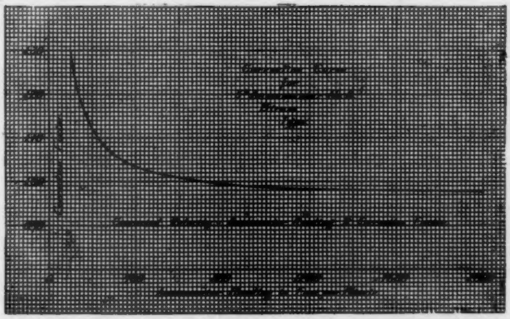

FIG. 1. CHART GIVING CORRECTION FACTOR CURVE FOR DETERMINING TRUE AIR-SPEED

to revolve at the desired speed. The next minute signal from the master clock circuit actuates a relay, the secondary circuit of which is in series with a magnet whose armature causes a toothed wheel to revolve one notch for every minute signal. Bearing on the side of this wheel is a metal brush which, upon the first minute signal, leaves an insulated section set in the wheel and comes in contact with the metal of the wheel itself. This closes another circuit, in series with which are two magnets. One magnet when thus energized starts the indicating mechanism of the anemometer. The armature of the magnet is attached through a cord to the lever which engages or disengages the indicating mechanism with the vane shaft. Simultaneously the other magnet starts the revolution counter. After the toothed wheel has rotated a predetermined number of notches, the brush meets another insulated section and the circuit through the anemometer and counter magnets is broken. The anemometer indicating mechanism is then disengaged by a spring fastened to the lever for engaging the indicating mechanism to the vane shaft. Simultaneously the revolution counter is stopped by the release of its magnet armature. The power is then shut off and when the arm is at rest the final anemometer and counter readings are recorded.

The results obtained with the whirling arm must be corrected for

air-swirl. The rotation of the arm and apparatus causes the air to be dragged along in the path of the anemometer. The effect of this air-swirl is to cause a reduction in the air-speed indicated by the anemometer, as the true speed of the instrument is of course with reference to the air through which it moves. Accordingly, in the determination of the true air-speed, the speed of the anemometer with reference to the ground as determined by the constants of the apparatus must be reduced by an amount which, although comparatively small, is too large to be neglected. The correction for air-swirl applied to the Bureau's whirling arm apparatus has been determined by the comparison of the results obtained when the same anemometers, similarly mounted, were calibrated first on the whirling arm in its circular path of travel and then in the straight air-stream of a wind tunnel.

DISCUSSION

MR. CHADWIN: In testing anemometers with the propeller wheel in a vertical position is there any difference in having this propeller blade or the propeller wheel in a horizontal or vertical position?

DR. BRIGGS: There certainly is a difference in the reading for the two positions. What would be really necessary would be a vertical wind tunnel with the anemometer in the position that it is going to be used ultimately and we are not equipped at present for such calibration.

W. F. STUTZ: Of course, the bearings would be designed for the anemometer operating in a horizontal position and in turning it you would have an entirely different state of affairs. The friction doubtless would be much greater.

P. NICHOLLS: Is the Bureau working upon and testing instruments for high velocities and large volume and low velocities and small volumes and areas?

DR. BRIGGS: We are now working on the question of anemometers for low air speeds and two methods are being tried out. One is a vane in the wind-stream which is deflected by the wind-stream and restored by means of a torsion wire. The other involves a very accurate gauge which will go back to the Pitot tube measurement, relying upon the precision of the gauge of the Pitot tube rather than the fan method.

F. R. STILL: Some years ago we depended entirely on the anemometer. Then we started investigating Pitot tubes and developed the tube that we use at the present time. But it seems to me that those interested in this subject should forget the anemometer and start off with some kind of a new instrument that is rugged enough and suitable to be easily carried around. There is a place for the use of the anemometer because when using the Pitot tube you need a certain amount of rigging.

MR. CHADWIN: In connection with the accuracy of the instruments we found it made a difference of as much as 2 per cent if the instrument is mounted on a board 2 in. thick instead of a plate ⅛ in. thick. Therefore if you want to get accurate results you have to be very careful to mount the instrument in as near as possible an undisturbable position.

No. 652

TESTS OF RADIATOR TRAPS

By W. F. Stutz[1], Washington, D. C.

Non-Member

THE traps considered are those used at the outlet of each radiator in the so-called vacuum system of heating in which low-pressure steam is circulated by maintaining a reduced pressure or vacuum in the return piping. The steam is brought to the radiators by one system of piping and the condensation and air carried off by a second

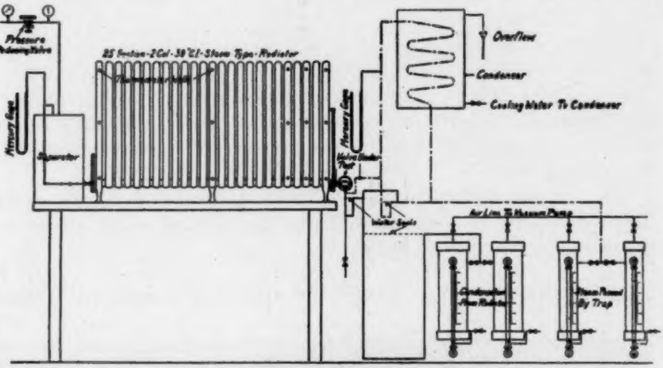

FIG. 1. DIAGRAM OF TEST APPARATUS

system. These traps, or valves, are placed in the return line at the outlet of each radiator, dividing the high pressure from the low pressure side of the system, for the purpose of discharging the condensation and air from the radiator and of preventing the escape of steam. They are a distinctive and essential part of a system of this character; the success of the system as a whole is principally dependent on the successful performance of the traps.

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[1] U. S. Bureau of Standards, Washington, D. C.

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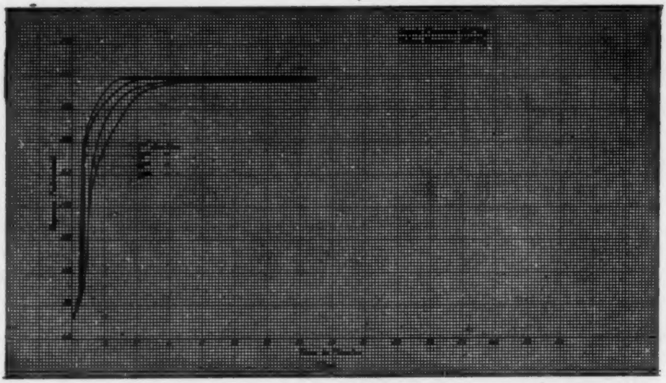


FIG. 2. SHOWING RAPID TEMPERATURE RISE IN ALL SECTIONS OF RADIATOR

The ideal operating performance for a trap for this purpose may be defined as follows:

1. It should keep the whole radiator at the temperature of steam supplied, by the continuous removal of condensation and air. The presence of water in the radiator is, also, in itself objectionable since any appreciable accumulation is likely to give rise to noisy operation.
2. It should prevent the wasting of any uncondensed steam to the return line.
3. It should be capable of quickly disposing of the relatively large amounts of condensation and air present when steam is first turned into the cold radiator.
4. It should without adjustment be capable of automatically adapting itself to the changes in operating conditions, relative to

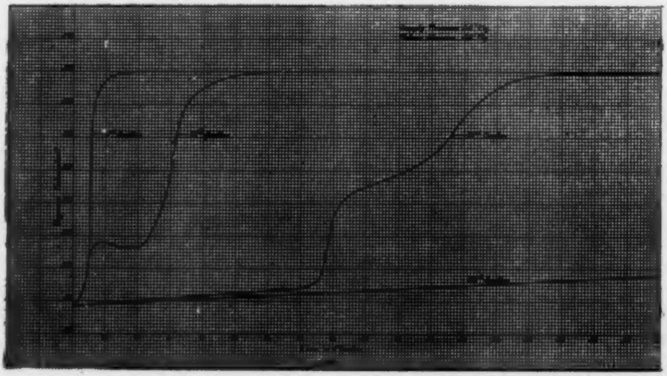


FIG. 3. SLOW DISTRIBUTION OF HEAT THROUGH RADIATOR

steam pressure and return line vacuum, which are likely to occur in practice.

The performance of the valve to meet the requirements listed can be determined by laboratory tests. There has, however, been no method developed for making an accelerated life test which satisfactorily reproduces actual service conditions.

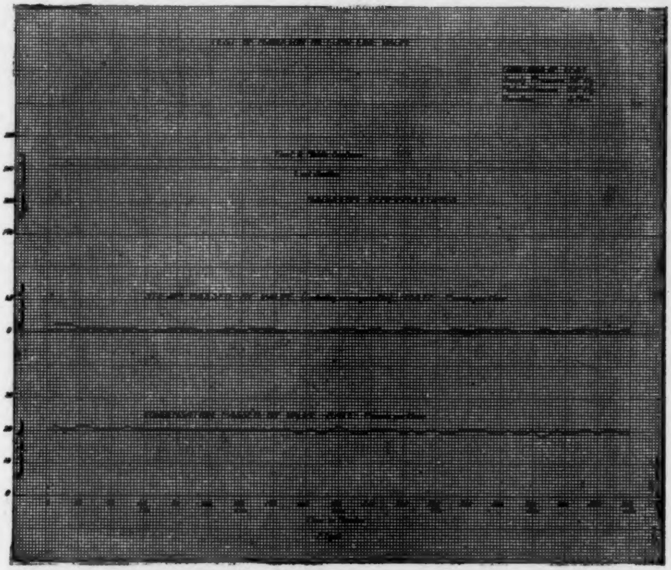


FIG. 4. A PRACTICALLY PERFECT VALVE PERFORMANCE

For the purpose of determining how closely traps in commercial production approximate the ideal performance defined, a series of tests was recently made at the Bureau of Standards of eight of the best known and most widely advertised traps, each represented by three identical stock samples.

Three operating tests under different conditions were made of each sample, as per the following Table 1:

TABLE 1. CONDITIONS OF TEST

Test No.	Supply Pressure	Return Line Vacuum	Duration Hours
	Inches of Mercury		
1	6	10	3
2	6	0	3
3	1	5	6

A further test was made in which steam was turned into the cold radiator and observations made of the time required for various sections of the radiator to reach the temperature of the steam supplied.

A diagram of the testing apparatus is shown in Fig. 1 which is largely self-explanatory. The radiator was of the steam type of 25 2-column 38 in. sections. The outlet section was of the drop-hub type fitted with an eccentric bushing. The condensation discharged by the valve was collected in either one of the two larger tanks. The vapor in the piping on the return side of the trap was condensed and collected in either one of the two smaller tanks. The tanks were installed in duplicate so that one could be cut out and emptied, while the second was in use, without

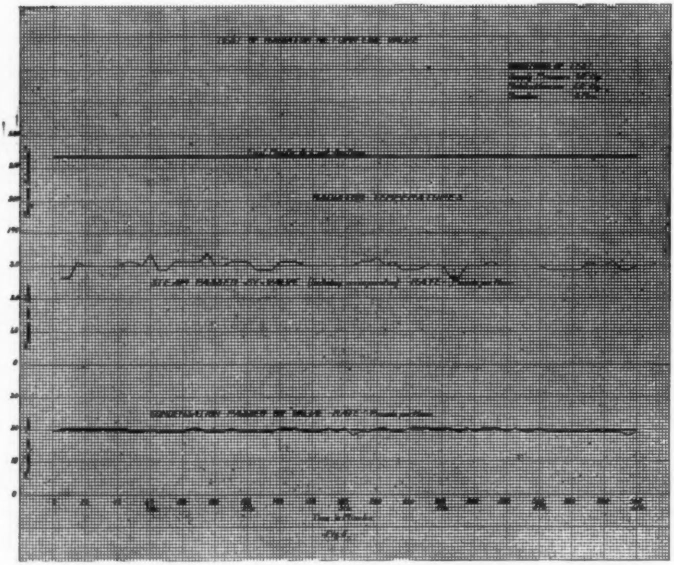


FIG. 5. VALVE WASTING EXCESSIVE AMOUNT OF STEAM TO RETURN LINE

interrupting the test. The gauges on the tank were graduated in pounds and fractions of a pound. The vapor pipe connection from the valve outlet to the condenser was made as short as possible (about 18 in. of ¾ in. pipe) and lagged. The riser to the condenser is drained through a water seal to the collection tanks. A gauge glass on the end section indicated visually the accumulation of water in the radiator.

Observation of the steam pressure, return line vacuum, and the temperatures in the radiator were made at 10 minute intervals. Observations of the rate at which condensation was discharged by the valve and of the rate at which vapor was condensed in the return piping were made at five minute intervals. The condensation collected in the vapor tank includes the steam passed by the valve as steam and also the re-evaporation in the return piping resulting from the discharge by the trap of water at a temperature higher than that corresponding to the pressure in the return line. In so far as the steam wastage obtained has not been cor-

rected for re-evaporation or for the compensating loss of heat from the portion of the vapor piping between the valve outlet and the point where it connects with the riser to the condenser, these results are comparative and are to be referred to the particular apparatus used.

The range in performance of the valves tested is illustrated by the typical curves shown in Figs. 2 to 6. Figs. 2 and 3 show graphically two extremes in the ability of different traps to bring the radiator quickly up to temperature when steam is first admitted to the cold radiator. Along the horizontal axis is plotted time in minutes from the starting of the tests. The curves show graphically the rate of rise in temperature

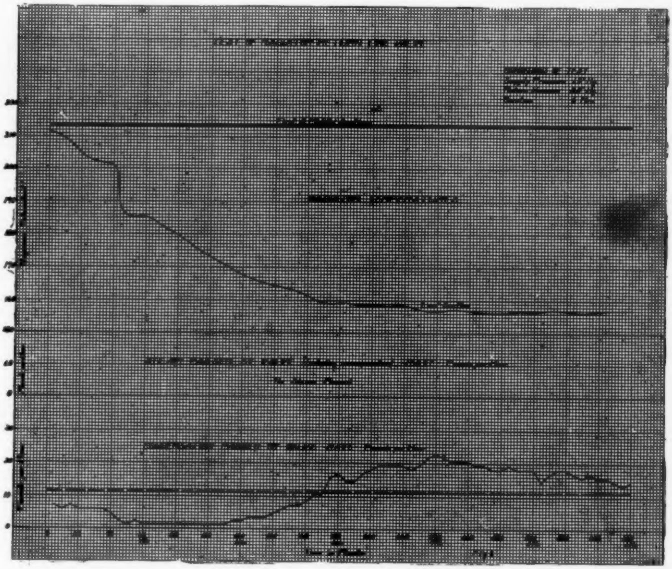


FIG. 6. STEAM PASSAGE INTERFERED WITH BY CONDENSATION AND AIR

in different sections of the radiator. In one case the whole radiator was brought practically up to the temperature of the steam supplied in about five minutes, in the second the last section had not reached 100 deg. in half an hour. Figs. 4, 5 and 6 show a graphic log of different typical operating performances over a six hour period. Simultaneous values of the rates at which the valve discharged condensation and passed steam to the return line and of the temperature in different sections of the radiator are plotted one above the other to a common time scale. Fig. 4 illustrates a performance which is practically perfect. The condensation was passed by the trap at a uniform rate, continuously as formed; the whole radiator was kept practically at the temperature of the entering steam; and the steam wastage was negligible. Fig. 5 shows the performance of a valve which discharges the condensation continuously as

formed, keeps the radiator up to the temperature of the entering steam but continuously wastes an excessive amount of steam to the return line. In Fig. 6 the opposite extreme of performance is illustrated. The trap wasted no steam to the return line but failed to keep the radiator cleared of condensation and air. The lower curve shows the trap passed only a fraction of the normal condensation (about 20 lb. per hour) for the first two hours, then at a gradually increasing rate for the next two hours until the normal rate was reached and again at a decreasing rate to the end of the test. The radiator was not cleared of water during the test period of six hours. The radiator temperature curves, above, show the resulting drop in temperature.

SUMMARY OF TEST RESULTS

A tabulation of the average results for each of the three performance tests of three identical stock samples of the eight commercial traps investigated showed the following:

1. Three traps showed an excellent performance which would meet a most exacting performance specification. All parts of the radiator were maintained at a temperature within 2 or 3 deg. of that of the entering steam; the condensation was discharged at a practically constant rate as formed; and the net steam wastage was well within a tolerance of 3 per cent of the steam condensed.
2. Two traps approximated the above performance but showed a comparatively large steam wastage; one trap showed a consistent excessive steam wastage and two consistently held back water in the radiator to a very marked degree, resulting in a decided temperature drop in the radiator.

DISCUSSION

C. F. EVELETH: Very frequently, by observing the products that are discharged from the trap, an attempt is made to determine whether the trap is a good one or a poor one. If you are going to attempt to analyze the different makes of traps to determine which is the most efficient you have got to depend upon some such evidence as has just been presented in this paper, rather than depend upon visibility alone. Of course, in the testing of radiator traps in the laboratory two very important considerations, which interest the owner, engineer and architect, as well as the contractor, are not disclosed. One of them is the operation of the trap under practical working conditions. That is, will a trap function properly when it operates in the presence of grease, oil, sand, scale, etc., which is continually accumulating in the radiator and has to be discharged from the trap? The other point is the probable life of the trap; how long will it last? I think that the engineers at large and also the owners are commencing to realize that there is something which determines the choice or selection of a trap besides the actual price. We oftentimes see a trap rated at 100, 200 or 300 sq. ft. without specifying under what conditions the trap will operate. The factor of safety is an-

other question. Is a trap rated up to its capacity? Does it have a sufficient factor of safety so that under unusual conditions, such as starting up the system when the room is very cold, with a very much greater rate of water discharge, it operates satisfactorily?

This paper shows clearly that the radiator traps with which we are familiar are divided into two classes. First, that class where you do not waste any steam at all but have a higher temperature drop from the first to the last section of the radiator, in which case you will not get the amount of heat expected. The second class is a type of trap which will waste quite a lot of steam and be inefficient in that way but will maintain all the sections of the radiator at the temperature corresponding to the operating steam pressure. One has efficiency one way and one has efficiency the other way.

C. W. STEWART: I think Mr. Still some time ago presented a synthetic air chart in which the air was penalized for certain qualities and I believe that the same thing could be applied to the thermostatic trap to determine upon an ideal condition. The trap should be rated at first with the efficiency with which it heats a radiator and be penalized a certain percentage for the pounds of steam it passes and also for the amount of water it turns back per minute.

THE PRESIDENT: I have personally given traps considerable study and it seems to me that we should separate the different functions of the trap. There are three distinct functions: First is the receding pressure; the second is what is the recession from the seat or working orifice; the third element is the question of durability. Simplicity may be a fourth factor, but I think most of us agree that the trap which has few sliding or moving parts or springs is better than the one that has many.

C. V. HAYNES: The thermostatic trap, to be what it should be, must be made from a metal that will not break, stretch or lose its tension, in operation, so that over a period of time the trap will function perfectly. Down at Annapolis, Md., a number of tests on traps for marine work are being conducted and I believe that this Society should conduct like tests from 0 lb. pressure up to 10 lb. pressure in order to get the carrying capacity of traps. The trap situation is one that the Society should take up and check up with Mr. Stutz and if the manufacturers of traps are not making a product up to a certain standard, the Society should set a standard which the manufacturer should equal or better if possible.

A. H. WOOLSTON: I can understand and appreciate that it would not be ethical to divulge the names of the various manufacturers of the traps which have been tested. However, I believe that it would be of great interest to the membership to know the types of traps represented in the tests and the performance of the two types. Thermostatic traps are divided into two groups or classes, one group in which the thermostatic member is located on the radiator side of the valve seat and the other group in which the thermostatic member is located on the vacuum side or on the discharge side of the seat.

J. A. DONNELLY: It is not a necessity from my point of view to put a steam trap in every room of a hotel. I think it belongs down in the cellar. I used to have an air valve in every room and found it very difficult to send the air down the return pipe with the water and remove it with an air valve in the cellar. Yet that problem has now been solved mechanically. Where is the most accessible place to locate a steam trap draining radiators? It is practical to put it in trenches with a piece of tile in front and I have found this to be one of the best and most accessible locations.

F. PAUL ANDERSON: This subject brings up perhaps the most interesting relationship of the Laboratory and the Society. It has been a well established principle that only those problems be considered that really deal with basic laws and keep away absolutely from those things that look like comparative tests. Now it seems to me the function of the Laboratory might be exercised in developing a standard method of testing a device like a trap and manufacturers might send to a laboratory traps to be tested under this standard method. There should, in my opinion, never be made at the Laboratory at Pittsburgh a series of comparative tests of traps or any other device that the members of this Society may be interested in. On the other hand, it seems to me that we are in a position to determine, through the help of all men interested in that particular device, a standard method of testing.

No. 653

CAPACITIES OF STEAM HEATING RISERS AS AFFECTED BY CRITICAL VELOCITY OF STEAM AND CONDENSATE MIXTURES

By F. C. Houghten[1], Pittsburgh, Pa. (Member)

and

L. Ebin[2], Pittsburgh, Pa. (Non-Member)

INTRODUCTION

It is intended that this shall be a complete report of the investigation thus far conducted of the critical velocity of steam and condensate mixtures in risers. This work, being carried on by the Research Laboratory of the American Society of Heating and Ventilating Engineers at the Pittsburgh Laboratory of the U. S. Bureau of Mines, was started in March, 1922, and a preliminary report was presented at the Semi-Annual Meeting in June, 1922 (Journal A. S. H.-V. E., Sept., 1922).

A program and outline of the problem to be followed and the results of a few preliminary tests were included in this report. Enough work had been done to discover some of the difficulties and form a few tentative conclusions. These difficulties have since been overcome in part and the conclusions in general verified.

Due to many years of work the boiler manufacturers have reached a point where heating boilers are well standardized. The design of, and heat emitted by, radiators are quite well known. However, when it comes to the flow of steam in heating pipes, particularly in one-pipe systems, there seems to be no standard and accurate data available which the heating engineer may safely use in his design of equipment. It was with the idea of obtaining as much information as possible regarding the flow of steam in heating pipes and of determining the factors affecting this flow that this problem was undertaken.

The plans to be carried out in connection with the outline of this investigation include the following problems, results of the first four of which are presented in this report.

[1,2] Research Engr., A. S. H.-V. E. Research Laboratory, U. S. Bureau of Mines.

Paper presented at the Annual Meeting of the American Society of Heating and Ventilating Engineers, Washington, D. C., January, 1923.

1. Critical Velocity of Risers—one and two-pipe systems.
2. Effect of rounded, reamed and unreamed entrances upon the critical velocity.
3. Effect of entrances of pipe cut with single and three-wheel cutters upon the critical velocity.
4. Effect of unions and couplings upon the critical velocity.
5. Critical velocity of mains pitching toward the boiler; both one and two-pipe systems.
6. Critical velocity of riser connections pitching back to the main; both one and two-pipe systems.
7. Critical velocity of horizontal pipes, all degrees of pitch.
8. Critical velocity of one-pipe radiator connections.

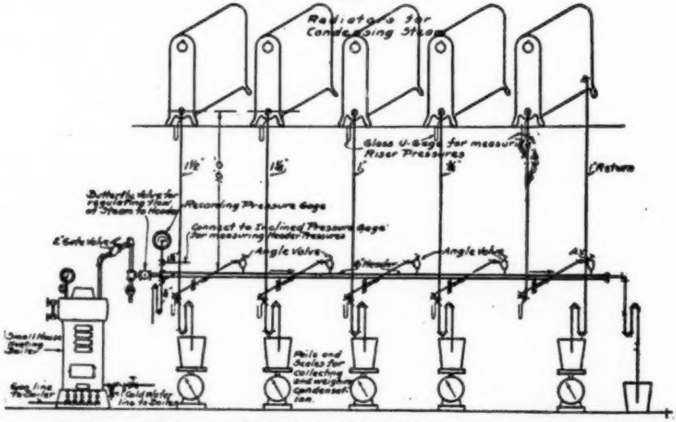


FIG. 1. APPARATUS FOR DETERMINING CAPACITIES OF STEAM HEATING RISERS AS AFFECTED BY CRITICAL VELOCITIES OF CONDENSATE MIXTURES

9. Critical velocity of first floor radiator connections.
10. Critical velocity of horizontal offsets in steam risers; both one and two-pipe systems.
11. Effect of globe valves in vertical and side positions upon the critical velocity.
12. Effect of angle valves for various lifts of the disc upon the critical velocity.
13. Influence of the starting load upon the critical velocity, also the effect of the size of the air valve orifice upon the starting load.

DESCRIPTION OF APPARATUS AND METHOD OF PROCEDURE

The apparatus, in general, is the same as that described in the previous report. A few changes have, however, been made in order to give greater accuracy or simplicity of operation, a diagram and a photograph of the apparatus used at present being shown respectively in Figs. 1 and 2.

Steam is raised in a house heating boiler and passes through a 2 in. gate valve and 2 in. butterfly valve into a 6 in. header. Five test risers, each 10 ft. long, are connected to the header by means of a 2 in. angle valve, two 2 in. street ells, a 2 in. union, a 2 in. reducing tee, reducing to the size of the riser, and a 2 in. horizontal connection 30 in. long. The size of the risers were varied from ¾ in. to 1½ in. The risers are connected at the top by a 2 in. ell, and a 2 in. nipple to the radiators.

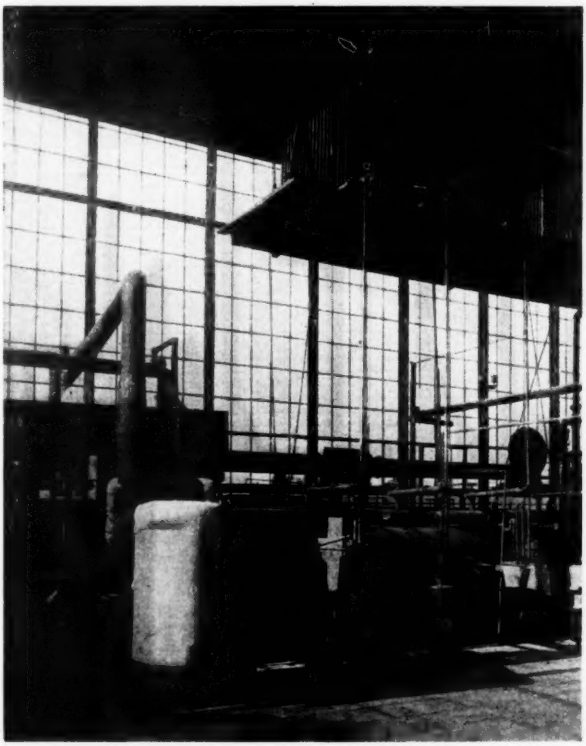


FIG. 2. APPARATUS FOR DETERMINING THE CRITICAL VELOCITY IN RISERS

The function of the radiator is merely to condense all the steam sent up the riser.

Four of the risers are connected as single pipe systems, that is, all condensation returns in the same riser in a direction opposite to the flow of steam. The fifth riser is arranged as a two-pipe system, the condensation returning by a separate line. All the condensation returns through a water seal into a bucket placed upon a spring balance. The length of a test varied from 15 to 30 min., depending upon the con-

sistency of the data obtained. Balance readings were taken at 3 min. intervals.

During the tests so far conducted, the radiators were operated at atmospheric pressure, the air vents being wide open. However, since the capacity of the radiator is greater than the capacity of the risers, no steam escaped from the air vent. The pressure of the steam in the header was kept constant for each test, by controlling the amount of steam allowed to enter the header by means of the butterfly valve. This pressure was measured by an inclined manometer sensitive to very small

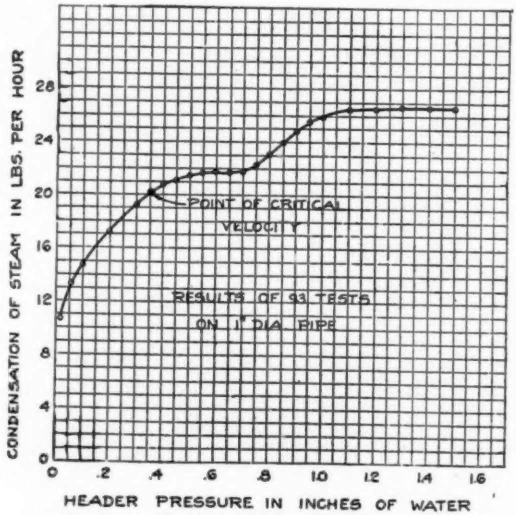


FIG. 3. RESULTS OF TESTS ON 1 IN. PIPE

pressure changes. To further avoid any fluctuations the pressure in the boiler was kept constantly at the blowoff point of the safety valve.

In the preliminary tests, made prior to publishing the first report, pressures in the header and top and bottom of the risers were measured by ordinary glass U-tubes. These values were determined to an accuracy of about 0.1 in. The curves presented in that report were plotted with total condensation in pounds per hour against pressure drop through the 10 ft. riser. As the pressure drop increased from zero to about 0.5 in., the capacity of the riser increased very rapidly and then approached a constant value at about 2 in. It was thought from the preliminary report the critical velocity probably occurs under 0.5 in., or at a point where increase in condensation with increase in pressure drop changes rapidly. With the methods then in use for measuring pressure, the pressure drop in the riser could not be determined with sufficient accuracy to bring out definitely this point. Hence any irregularity in the curve, which might be expected in this region, could not be detected.

Early in the investigation it became apparent that in order to determine the critical point on the curves and therefore to solve the problem some means must be found to determine more accurately this part of the curve. A solution was sought in two directions. First, the adoption of some means of determining pressures at the bottom and top of the risers to an accuracy of at least 0.01 in. of water column. Second, the discovery of some other measurable factor which would vary with the amount of steam passing through the risers.

Considerable time was spent in trying to devise some means of accurately determining riser pressures. The chief difficulty in making this determination is the passage of water from the riser into the tube

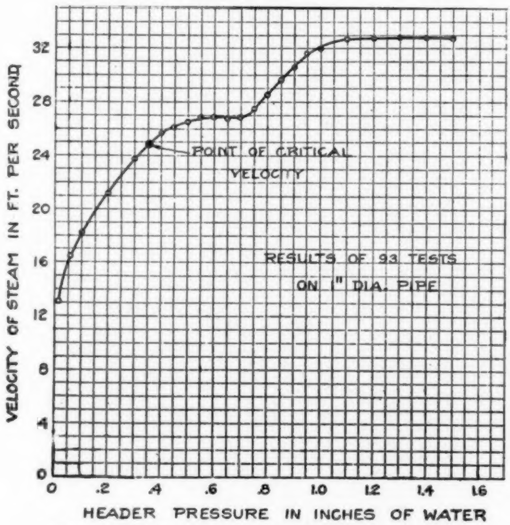


FIG. 4. RESULTS OF TESTS ON 1 IN. PIPE

leading to the pressure gauge. It can be seen that if a gauge is to register pressures to a 0.01 in., and a single drop of water finds its way into the gage connection, sealing off the tube as it will by capillary attraction, the static pressure produced by this single drop of water may be many times greater than the pressure that is being measured.

While many devices were tried and many suggestions from persons interested in this question were received and tried, no solution of this phase of the problem has yet been found. The accuracy with which these pressures may be measured has, however, been increased to about 0.05 in. of water column.

A solution has been found, however, in the second direction. The header pressure while not directly proportional to the drop in riser pressure, does vary with it and therefore with the rate of condensation.

TABLE 1. SUMMARY OF TESTS ON 1 IN. DIAMETER PIPE

Tests 103 to 239 Inclusive

Inside Diameter of Pipe = 1.049
Area of Pipe = 0.864

Header in. of water pressure	lb. per hr. Condensation	Vel. of steam ft. per sec.	Test No.
0.02	10.50		106
	10.75		122
Av.	10.63	13.1	
0.06	12.50		107
	13.75		153
	13.80		160
Av.	13.35	16.43	
0.1	14.50		105
	15.00		138
Av.	14.75	18.18	
0.2	16.75		104
	17.50		120
	17.50		121
	17.00		210
Av.	17.19	21.22	
0.3	19.75		103
	18.75		123
	19.25		134
	20.13		207
	18.75		220
	19.00		228
Av.	19.27	23.73	
0.4	20.75		108
	20.75		128
	21.63		148
	20.50		204
	20.50		211
Av.	20.83	25.75	
0.45	21.13		208
Av.	21.13	26.10	
0.5	21.50		109
	21.25		113
	22.38		124
	21.25		131
	20.88		133
	21.56		135
	21.38		218
Av.	21.46	26.51	
0.55	21.75		150
Av.	21.75	26.86	
0.6	21.60		110
	21.50		112
	22.50		127
	21.75		144
	21.78		201
	21.75		212
	21.63		221
	21.50		229
Av.	21.75	26.80	
0.65	21.75		152
	21.38		219
	21.67		236
Av.	21.60	26.68	
0.7	21.86		114
	22.50		118
	21.75		125
	22.00		126

Inside Diameter of Pipe = 1.049
Area of Pipe = 0.864

Header pressure in. of water	Condensation lb. per hr.	Vel. of steam ft. per sec.	Test No.
	21.67		136
	21.75		205
	21.50		217
	21.50		226
	21.50		233
Av.	21.78	26.90	
0.75	22.50		139
	22.25		146
	22.50		154
	21.88		238
Av.	22.28	27.51	
0.8	23.50		117
	22.75		129
	23.25		203
	22.50		213
	23.75		222
	22.75		235
Av.	23.08	28.50	
0.85	24.12		140
	24.26		156
	23.50		227
Av.	23.96	29.59	
0.9	25.50		115
	25.00		132
	24.63		216
	24.13		230
Av.	24.81	30.64	
0.95	25.75		149
	25.50		158
Av.	25.63	31.65	
1.0	25.42		116
	26.00		119
	25.50		130
	25.63		137
	26.50		141
	25.75		202
	25.75		214
	25.75		234
Av.	25.78	31.85	
1.1	26.50		147
	26.50		151
	21.38		237
Av.	26.46	32.67	
1.2	26.50		142
	26.67		143
	26.25		224
	26.50		231
Av.	26.48	32.70	
1.3	26.88		145
	26.63		155
	26.38		206
	26.75		239
Av.	26.66	32.93	
1.4	26.63		157
Av.	26.63	32.89	
1.5	26.88		159
	26.25		232
Av.	26.56	32.80	

Without serious difficulty the header pressure can be measured to 0.01 in. by means of a special inclined gauge using Xylol.

By now plotting condensation per hour against header pressure, determined to 0.01 in. accuracy, the important part of the curve becomes much stretched out and an attempt can now be made to study it more carefully and to note what is happening.

CRITICAL VELOCITY OF ONE-PIPE RISERS

After running a few tests in which the header pressure was accurately determined by means of the sensitive gauge mentioned before and plotting

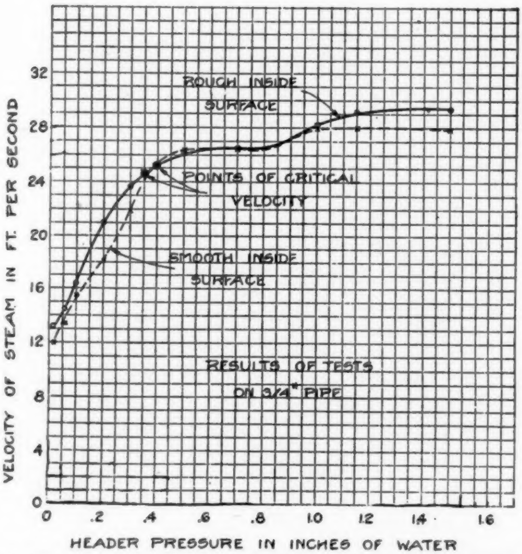


FIG. 5. TESTS OF ¾ IN. PIPE WITH ROUGH AND SMOOTH INSIDE SURFACES

these points on a larger scale than heretofore, a wave or an abrupt change in the duration of the curve was found in that part of the curve in which it had been expected to find the critical velocity.

Accordingly, a very exhaustive series of tests was made on one of the 1 in. risers in order to definitely establish this part of the curve. Ninety-three tests were made at pressures under 2 in.

Table 1 is a summary of these tests. The tests in this table are numbered in the order in which they were made. It will be seen that tests at various pressures were repeated in irregular order. This was done in order to make certain that no factor of time or order of test would affect the results. To make certain that the irregularity was not due to some unknown cause the connections between the header and

radiator were taken down, examined and reassembled on three different occasions. In Figs. 3 and 4 condensation in pounds per hour and velocity in feet per second, respectively, are plotted against header pressure for this series of tests. Comparing these curves with the corresponding curves for the 1 in. pipe as given in the previous report it will be seen that the portion of the curve below 1½ in. pressure is much stretched out along the pressure axis. Also there is a marked wave in the curves between header pressures of 0.4 in. and 1.1 in. At a header pressure of 0.4 in. the curves bend sharply to the right, indicating that the capacity

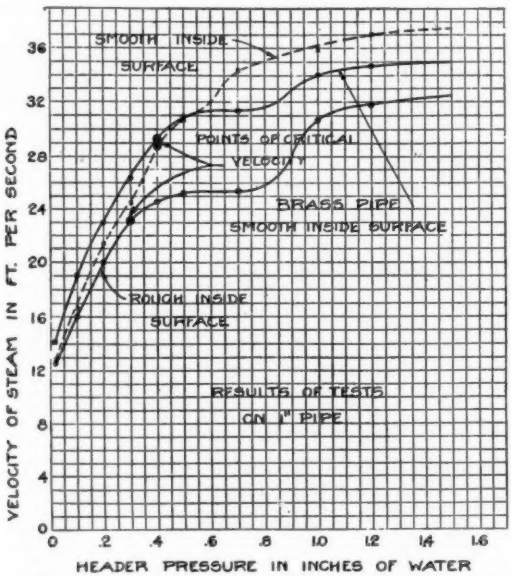


FIG. 6. TESTS OF 1 IN. BRASS PIPE ALSO PIPE WITH ROUGH AND SMOOTH INSIDE SURFACES

of the pipe for carrying steam, or the velocity of the steam, ceases to increase as rapidly with the same increment of pressure. In fact there is little increase in condensation or velocity until a header pressure of 0.7 in. is reached, when it again increases very rapidly.

The point at 0.4 in. header pressure where the direction of the curve changes abruptly represents a critical point on the curve, that is, a point where the laws governing the flow of steam in the pipe changes. The fact that this critical point on the curve takes place so near where we had, for reasons mentioned in the preliminary report, expected to find the critical velocity leaves no doubt that this is the point sought. We have therefore, solved the first phase of the problem, that is, we have determined a means of finding the critical velocity.

The next phase of the problem is to determine the critical velocity for various sizes of pipe.

In carrying on this investigation it soon became apparent that there is considerable variation in internal diameter and smoothness of pipe as manufactured. A variation of as much as 0.05 in. in internal diameter has been found in various pipes tested. This, however, does not offer a serious problem since all pipes were measured and the velocity determined from the capacity and actual diameter. Variation in roughness of pipe used for test does, however, offer a problem to be overcome in order

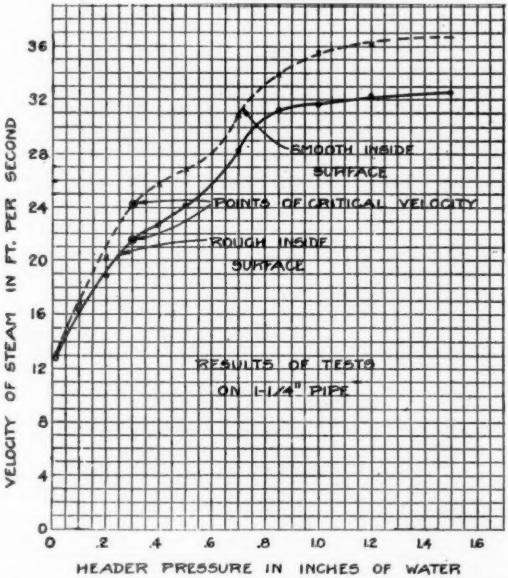


FIG. 7. TESTS OF 1¼ IN. PIPE WITH ROUGH AND SMOOTH INSIDE SURFACES

to obtain data which will be applicable to the average run of pipe. It was thought desirable to run tests on at least two samples of pipe of each size, one smoother and the other rougher than the average. Permission was obtained to go into the warehouse of one of the large pipe manufacturers and choose samples from their entire stock. After examining hundreds of pipes, two samples of each size, one rough and one smooth, were chosen.

When the pipe was all received and carefully inspected, it was found that there was considerable variation in smoothness between the two kinds of pipe for the 1 in. and 1¼ in. diameter. However, for the ¾ in. and 1½ in. diameter there was very little variation in smoothness between the two pipes. In addition a brass pipe of 1 in. diameter was chosen of supposedly very smooth inside surface.

Tests were run upon these nine pipes for the following purposes:

1. To determine the critical velocity for pipes of various sizes.
2. To determine the effect of the smoothness of the pipe upon the critical velocity.
3. To determine the variation in capacity due to the variation in size and smoothness of pipe upon the market.

The results of the tests upon these nine pipes are plotted in Figs. 5, 6, 7 and 8. In each case the velocity in feet per second is plotted against the header pressure. For the 1 in. pipe, results are plotted for the rough

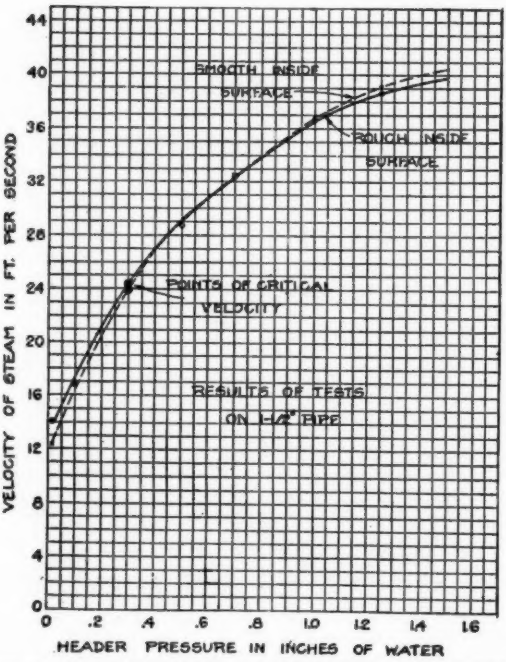


FIG. 8. TESTS OF 1½ IN. PIPE WITH ROUGH AND SMOOTH INSIDE SURFACES

pipe, smooth pipe and brass pipe. In the other three cases the results are plotted for the rough and smooth pipe.

On analyzing these curves the following points seem of significance:

1. The smaller the size of the pipe and thus the smaller the internal area of the pipe, the more pronounced is the wave portion. As the size of the pipe increases the wave becomes less pronounced. For the 1½ in. pipe it practically disappears. However, a distinct change in slope from the vertical towards the horizontal can still be found. If tests were run at very small increments of pressure

change, 0.01 in. of water instead of 0.10 in.; it may be that the wave portion would show much more clearly for larger size pipes.

2. If the results of the rough pipe and smooth pipe are compared this fact stands out. In the two cases where there is considerable difference in internal smoothness of the pipe as for the 1 in. and 1¼ in. pipes, there is considerable difference also in the value obtained for the point of critical velocity. On the other hand, for the two pipes where there is apparently very little difference in the internal smoothness such as the ¾ in. and 1½ in. pipe there is also very little difference in the value obtained for the critical velocity of the two pipes. Furthermore, if the 1 in. smooth pipe is compared with the 1 in. brass pipe, which was of exceptional internal smoothness, it will be noted that not only the critical velocity for the two pipes is practically the same but that it is considerably greater than the critical velocity for the 1 in. rough pipe. From this it is proper to believe that the critical velocity is affected somewhat by the internal smoothness of the pipe. It would seem more logical, however, to believe that the critical velocity is independent of the smoothness and that it is the effective area which varies with the smoothness. Since the velocity is determined by using the actual area, any change in effective area with smoothness would make the velocity appear to vary with the smoothness.

3. The point of critical velocity for all tests ranging from a very rough to a very smooth pipe varies from 22.5 ft. per sec. to 29 ft. per sec. A very good average for the entire set of tests would be 25.0 ft. per sec. The maximum variation in capacity of the various sizes of pipes tested is given in Table 2.

4. The header pressure at which the critical velocity was found in all the tests on pipe ranging from rough to smooth varies from 0.30 in. of water to 0.40 in.

5. In view of 3 and 4 it may be said that for the sizes of pipe tested and within the limits of experimental error the critical velocity is independent of the size.

EFFECT OF SHAPE OF ENTRANCE UPON CRITICAL VELOCITY

Tests were then continued upon the apparatus for the purpose of determining the influence of certain factors on the critical velocity. The first of the factors to be investigated was the effect of the shape of the entrance.

In these tests all 1 in. risers were used and the entrance of each riser was cut in a different manner. Fig. 9 shows the four types of entrances tested. Sketch *a* shows the entrances squared, *b* shows a reamed entrance, *c* a rounded entrance and *d* an entrance cut with a single or three wheel cutter. Type *d* was not reamed nor altered in any way and the burr made by the cutting tool left intact. Sketch *d* shows the approximate shape of the burr left by the cutters.

The results obtained in these tests are plotted in Fig. 10. The capacity in pounds per hour is plotted against drop in riser pressure in inches

of water. The curves for the reamed, rounded and squared entrances are each the average of tests on four different pipes. In the case of the signal and three wheel cutters only one pipe of each type was tested. These tests were made before the more accurate method of measuring header pressures was adopted. Hence, the capacity of the risers at the different critical points cannot be compared, but their capacities at the maximum points of the curves can be compared.

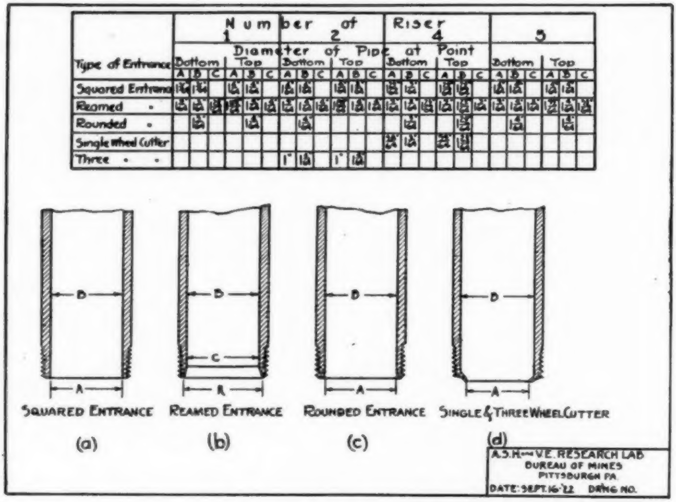


FIG. 9. FOUR TYPES OF ENTRANCES TESTED

The maximum capacity obtained in these tests are as follows:

		Per cent decrease
Reamed entrances..........	24.7 lb. per hr.	0
Rounded entrances.........	23.9 lb. per hr.	3.2
Squared entrances..........	22.2 lb. per hr.	10.1
Three-wheel cutter	19.2 lb. per hr.	22.2
Single wheel cutter.........	17.6 lb. per hr.	28.7

As was expected, the results show a great capacity for the reamed type, slightly smaller for the rounded pipe and so on. Using the reamed pipe as a standard of comparison, it is found that the rounded entrance shows a decrease of 3.2 per cent, the squared entrance a decrease of 10.1 per cent. Furthermore, if the burr left by a cutter is not cleaned out by reaming, the effect is a reduction of from 20 per cent to 30 per cent in the capacity of that pipe.

While the results indicate a slight difference between the single and three wheel cutters, due to the insufficiency of the data obtained, there is no attempt here to designate the one cutter as giving a higher capacity than the other. It may be stated, however, that any cutting tool which

leaves a burr in the entrance of the pipe will reduce the capacity of that pipe as much as 30 per cent. The amount of reduction depends upon the amount of burr made by the cutter. From a practical viewpoint, since any burr will decrease the capacity, care should be taken that all burr is removed from the pipe and furthermore that all pipe is reamed. This fact has been well known, nevertheless the results of these tests bring it out conclusively.

EFFECT OF UNIONS AND COUPLINGS

A series of tests were conducted to note the effect of unions and couplings upon the critical capacity. Unions or couplings were inserted

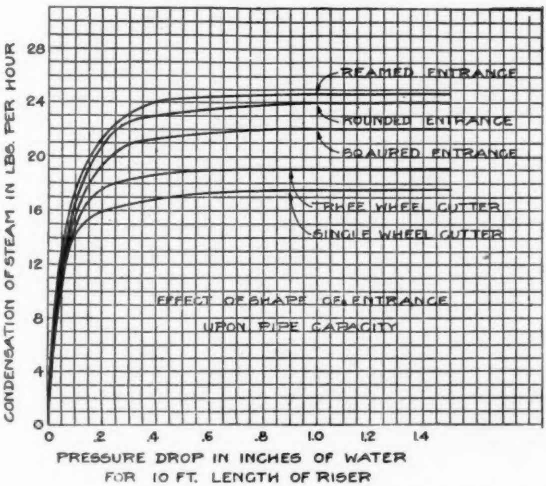


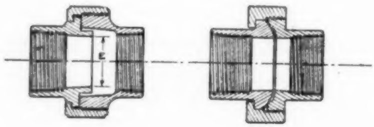
FIG. 10. RESULTS OF FOUR TYPES OF ENTRANCES TESTED

in the middle of a 1 in. riser. Both a lip joint and a ball joint union were tested, sectional diagrams of these two unions being shown in Fig. 11.

The results of these tests indicate that there is no apparent changes in the capacities due to the union or coupling in the line. However, it is essential to bring out this point. The results of the first tests upon the lip joint unions were slightly lower in capacity than for plain pipe. On taking down the unions and examining them carefully it was discovered that the diameter of entrance E (indicated in the sketch) was a trifle under size. This entrance was then reamed out and tests continued. The capacity returned to the normal value for that pipe. This seems to indicate that not only is it possible to have a constriction in certain types of unions, but also that the capacity of the riser is determined by that constriction. This is further verified by the diminished capacities accompanying the constrictions left by cutting tools.

EFFECT OF ORIFICES UPON THE CAPACITY

A series of tests were run to determine the extent to which short constrictions in a pipe will decrease the capacity of the system. Orifices of 1 in., ¾ in., ½ in. and ¼ in. diameter were inserted in the union



LIP JOINT UNION BALL JOINT UNION

FIG. 11. TYPES OF UNIONS TESTED

in the center of the 1 in. risers. The results of these tests are indicated by points *A, B, C* and *D* in Fig. 14 and will be discussed later.

EFFECT OF VARIATION IN PIPE SIZES UPON THE CAPACITY

It is significant to note the variation in the capacity of pipe upon the market that may be due to either variation in size or internal smoothness. As brought out before, the variation in size alone may be 0.05 in. in diameter or even more. The variation in internal smoothness may be very great.

TABLE 2. EFFECT OF SMOOTH AND ROUGH PIPE

Per cent difference due to Variation of Pipe

	Maximum Condensation, lb. per hr.			
Capacity of Pipe	¾ in.	1 in.	1¼ in.	1½ in.
Minimum	14.00	24.89	45.42	70.50
Maximum	15.20	30.08	52.08	82.00
Per Cent Variation	8.6	20.8	14.7	16.3

Table 2 shows the maximum variation in capacity found for the various sizes of pipes. It can be seen that there may be a variation of as much as 20 per cent in capacity due to the variation in size and smoothness.

GENERAL DISCUSSION

The curves drawn from the average results of all tests are shown in Figs. 12 and 13. In Fig. 12 velocity is plotted against header pressure. It can be seen that the points of critical velocity for the four curves all fall within a small region and vary from a velocity of 22.8 ft. to 25.2 ft. per sec. It will also be noticed that the maximum velocity increases with the size of the pipe. In Fig. 13 the condensation in pounds per hour is plotted against the header pressure. The capacity of the different sized pipe for carrying steam are all plotted on the same scale and for

this reason the wave is not so pronounced in the curve for the smaller pipe.

The maximum capacity of the pipe for carrying steam and condensate mixtures is plotted against the inside area of the pipe in square inches as shown by Fig. 14. The points of maximum capacity for this curve were obtained from the flat portion of the capacity-header pressure curves in Fig. 13. This curve indicates that the maximum capacity varies greatly with the size of the pipe.

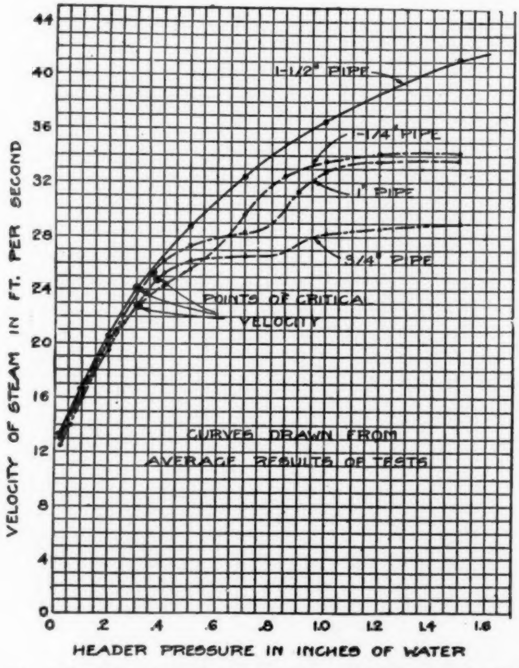


FIG. 12. CURVES DRAWN FROM AVERAGE RESULTS OF TESTS

Points *A, B, C* and *D* represent the maximum capacities obtained with the 1 in., ¾in., ½ in. and ¼ in. orifices inserted in a union in the 1 in. pipe as mentioned before. As can be seen, points *A, C* and *D* all fall very close to the maximum capacity curve. However, for some reason not discovered, point *B* for the ¾ in. diameter orifice fell somewhat lower than was expected.

The results obtained for the maximum capacity of the 1 in. pipe cut by the one- and three-wheel cutters and not reamed are also plotted against the free area of the pipes left at the constrictions. These points fall close to the maximum capacity curve.

This curve shows conclusively that the effect of any constriction in a pipe, no matter how short the constriction, is to reduce the capacity of the pipe to that of a pipe equal in internal diameter to the free area of the constriction.

A second curve in Fig. 14 gives the capacity of the pipe at the critical velocity for any area. This curve is found to be a straight line. Furthermore, at very small pipe areas the maximum capacity and critical capacity curves coincide. This would indicate that for very small sizes of pipe

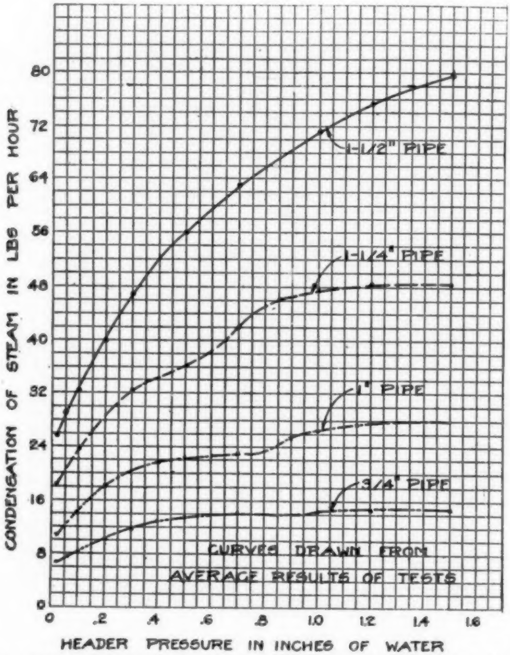


FIG. 13. CURVES DRAWN FROM AVERAGE RESULTS OF TESTS

the critical velocity and maximum velocity are practically the same. This is borne out by the curves in Fig. 13.

The primary object of this investigation has been to determine sizes of risers allowable for carrying steam, with the counter flow of condensate, without serious interference. By serious interference is meant intermittent flow, surging, water hammer, or any other departure from smooth operation. The work so far reported has been necessary to the end of obtaining such data. The data itself is given in Table 3. The values in the first column are based on 18 ft. per sec., which allows a factor of safety of approximately 20 per cent. The values in the second column are based on a velocity of 22 ft. per sec. or the minimum velocity found

for any pipe tested. The values for the 2 in. pipe were not determined by test, but are based on the conclusion drawn from the tests that the critical velocity for the 2 in. pipe is the same as that for the pipes tested.

TABLE 3. CAPACITIES OF ONE PIPE RISERS

Diam. of Pipe in.	Cond. lb. hr.	B.t.u. loss hr.	Sq. ft. Rad. Based on 240 B.t.u.	Cond. lb. hr.	B.t.u. loss hr.	Sq. ft. Rad. Based on 240 B.t.u.
¾	8.9	8640	36.0	10.9	10580	44.1
1	14.5	14070	58.6	17.8	17270	72.0
1¼	25.2	24460	101.9	30.8	29890	124.5
1½	34.2	33190	138.1	41.8	405.60	169.0
2	56.5	54830	228.4	69.0	66960	279.0

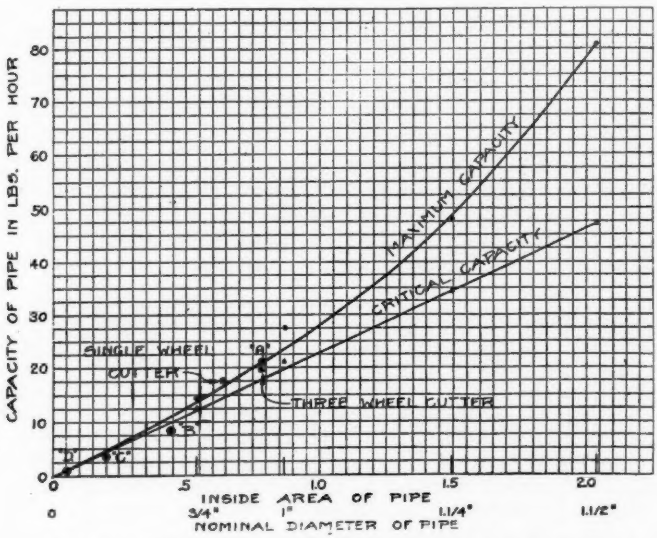


FIG. 14. CURVES SHOWING EFFECT OF CONSTRUCTIONS IN VARIOUS SIZES OF PIPE

The table gives the capacity of the pipe in pounds of steam per hour, B.t.u. transmitted per hour, and square feet of radiation based on 240 B.t.u. per sq. ft.

RECOMMENDATIONS

The investigation has brought out the following outstanding points which seem of sufficient importance to the heating trade to be summarized:

1. All pipe should be reamed. Pipe cut by an ordinary wheel cutter and not reamed will decrease the capacity of a one-pipe system by as much as 30 per cent.

2. Pipe with any unusually large bump, blister, or other constriction on the inside should not be used. The capacity of a one-pipe system is no greater than the capacity of the free area left by its greatest constriction.

3. Unions and all other fittings entering into a riser should be of full size opening.

4. The size of pipe for one-pipe risers carrying any quantity of steam should be determined in accordance with column one of Table 3, or by column two after applying the desired factor of safety.

ACKNOWLEDGMENT

The writers are indebted to Carnegie Institute of Technology for the services of Prof. L. S. O'Bannon, who spent three months at the Laboratory, and also to J. A. Donnelly, C. V. Haynes and Prof. S. E. Dibble, for their co-operation.

DISCUSSION

T. WEINSHANK: The point of critical velocity to the next point where there is a critical rise is the period where a capillary attraction for a peculiar state of the water exists and therefore it continues. How do you explain that the point of critical velocity is the same in the rough 1¼ in. pipe as in the smooth?

F. C. HOUGHTEN: We looked over hundreds of pieces of pipe of that size and picked out a rough pipe and a smooth pipe. There was, however, little variation. The results given are the minimum capacities for any pipe tested.

L. A. HARDING: I would suggest that a definition of what is meant by critical velocity as applied to this particular problem be given in this paper. There is a shade of difference in the meaning between the critical velocities as we understand them or as they are used in the flow of fluids in pipe and as applied to this problem.

H. M. HART: Sometimes we are able to install a riser with just a vertical pipe, but frequently it has to be offset. The common practice is to offset with 45 deg. elbows and I think that this condition ought to be taken into consideration before we go too far in taking these results as safe for all conditions of installation.

J. A. DONNELLY: I am rather of the opinion, from many years' experience, that critical velocities are more serious in the smaller pipes than in the larger. That is, when you get into critical velocities in smaller pipes you get an entire stoppage; at least you get such a failure of heating that you have scarcely any. But when you get into larger pipes and get water in them, very often you can maintain your velocities with considerable drop of pressure and blow the obstruction out of the way. I think we find that same thing in two-pipe risers.

My belief is that we might modify our two-pipe-riser practice to keep below the critical velocities in smaller sizes, 1¼ and 2 in. We do not have to pay any attention to critical velocities in two-pipe risers when we get up to 2 in. and larger. In larger pipes pitching back against the flow of steam may be entirely different.

H. M. HART: The flow of steam through orifices has nothing to do with the critical velocity in a one-pipe heating system. This fact does not apply to the two-pipe system or the flow of steam through orifices.

W. H. CARRIER: There are two types of research and engineering. The Germans always figure on theory and disregard the practice. I think we are inclined to figure on practice and disregard theory. The proper thing is a combination of both and we have come to it in this paper. Theory, if everything is taken into consideration, must agree with practice; if it does not agree with practice there is something wrong with the theory.

In engineering we have a broad field. An engineer has got a lot to learn. He cannot keep it all in his head. But if he can have a general broad principle, he can apply it to a hundred different applications of similar nature. That is its value to the general engineering profession in the discussion of the relation between the theory and the practice. Instead of having a variety of formulae or relationships you would need to know only one.

T. WEINSHANK: Practice is the application of theory. Practice is also the result of theory. In my judgment the Research Laboratory can and will check up our practice if same oversteps the bounds of theory. If the Research Laboratory can demonstrate to us that our practice is founded upon theory, then the Laboratory has accomplished a good deal.

If the Research Laboratory had not done a thing since its infancy other than to bring out Table 3 and its conclusions, all the money we have spent on it would be well justified.

P. J. DOUGHERTY: The critical velocity discussed in this paper is what we call a mechanical cause. In effect it depends upon gravity bringing the water down. In other words, we have an application of force; the momentum of the steam going up with the force of gravity drawing the temperature of the water down. When those two forces begin to balance that is when you get your critical velocity. Critical velocity, as ordinarily understood in air flow, is the point where stratification ceases and turbulency begins.

A theoretical formula applies to ideal conditions and naturally incorporates those ideal factors. A practical formula incorporates those other variables. They never duplicate. The only differences there are today between theory and practice is that theory provides for a precise number of factors, like in an algebraic equation. There is a precise set of factors and a precise answer. Now, therefore, if we change any of these, which is always a case in practice, then you are outside the realm of

theory. The practitioner allows for the factors that the theorist does not include in his equation.

C. F. EVELETH: Everybody who has ever had anything to do with the question of heating calculations has realized the lack of definite information. The papers which have been presented to the Society by the Research Laboratory have begun to establish some of the fundamentals; they have begun to give us something that we can tie to; they have given us something from which we can start from and work up. And, when we have the fundamentals, the factor of ignorance or the factor of safety can be very materially reduced.

In the past we have had to apply factors of safety, because there were a great many things taking place of which we knew nothing, and we had to discount those things by applying this factor. It seems to me that this paper marks a very distinct progressive step in our knowledge of what takes place inside of pipes. Every one of us has been up against the problem of risers and mains which, according to the rules which we had, ought to give satisfactory results, but for some reason or other did not. The more information of this character that we have, the greater is going to be our confidence in going out and tackling the new problems. I want to commend the authors of this paper for their excellent and painstaking work, particularly the list of recommendations.

Many papers do not include definite applications and go off into the realms of theory or pure mathematics. They go over our heads and we do not appreciate their value. Here is a case where we have something that is going to assist everyone of us in our daily work in connection with the figuring of pipe sizes. I feel very sure that as time goes on and the Research Laboratory has opportunity to investigate this subject more thoroughly, a good many of our troubles along the subject of pipe sizes will be eliminated.

F. PAUL ANDERSON: The function of the Laboratory finally should be, to give to the engineering profession a basis of theory. We want to make the Laboratory of some practical benefit all the time. It would be unsatisfactory, I take it, to present here mathematical theory all the time. On the other hand, it would be very unsatisfactory to present simply a lot of observations without any conclusion. We are trying to do both. We must adhere to theory, and the higher we develop our theory the better our practice will be.

F. D. MENSING: I think we could avoid a great deal of discussion if we had a system of definitions. I have noted in all our talks that we use words and do not seem to be talking exactly in the same language. It might be a good idea at some future time to form a system of definitions which would avoid this trouble.

No. 654

SOME PHYSIOLOGICAL REACTIONS TO HIGH TEMPERATURES AND HUMIDITIES

By W. J. McConnell[1], Pittsburgh, Pa. (Non-Member)
and
F. C. Houghten[2], Pittsburgh, Pa. (Member)

INTRODUCTION

It is generally acknowledged that heterothermic, or cold-blooded animals are directly influenced by the water or air in which they live; and it is likewise conceded that homeothermic, or warm-blooded animals, in which group man belongs, are physiologically affected by changes of external temperature; but notwithstanding the fund of varied information on the subject, a comprehensive account of the action that various temperatures and humidities have on health and comfort are still largely a matter of conjecture.

Our conception of the relationship between human beings and the atmosphere in which they live; a review of the progress accomplished in this field of research work; and a plan of the present study undertaken by the three services—namely, the Research Laboratory of this Society, the U. S. Bureau of Mines, and the U. S. Public Health Service—have been presented in previous papers.

Progress reports of this co-operative study have been submitted in like manner. One year of fruitful work has terminated and the results of the experiments so far finished are here recorded for the benefit of those interested.

SCOPE OF WORK

Before entering into a discussion of the physiological findings of our experiments, a brief statement of what has been attained in the present study and a general reference to future work contemplated, together with a description of facilities available and methods used, may serve to elucidate certain aspects of the study.

The work is divided into four phases, and each phase presents a definite problem, as follows:

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[1] Passed Asst. Surgeon (R) United States Public Health Service.
[2] Research Engr. A.S.H.-V.E. Research Laboratory, U. S. Bureau of Mines.
Paper presented at the Annual Meeting of the American Society of Heating and Ventilating Engineers, Washington, D. C., January, 1923.

1. The first pertains to a study of the physiological effects of various temperatures and humidities on human subjects at rest in still air. This phase of the work is practically completed, and the results of the experiments above 90 deg. fahr. are the subject of this paper.

2. It is proposed to study the physiological reactions to different temperatures and humidities with various air velocities when the subjects are at rest.

FIG. 1. POTENTIOMETER; RECORDING BODY TEMPERATURES

3. A study likewise will be made when the subjects are performing either physical or mental work.

4. A study of the changes in these reactions while the subjects are at work will be made in still air.

EQUIPMENT AND METHODS

The equipment at the Laboratory for the study of these problems is complete. Two fully equipped chambers, insulated by cork board, were designed to maintain air conditions automatically at the desired temperature and humidity. The temperature, humidity and air motion of each room may be controlled independently of one another.

The air conditions are controlled by apparatus outside of the chambers, and entirely separated from them. Instruments for observing the body and surface temperatures of the subjects, also for recording the rate of respiration and of the heart pulsations, and the apparatus for basal metabolism work are located in an adjoining room.

The oral temperature of each subject is registered by the one-minute clinical thermometer, while the rectal and surface temperatures

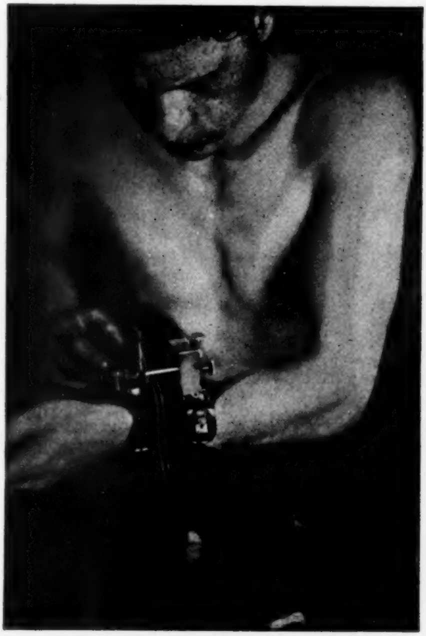

FIG. 2. INSTRUMENT FOR AUTOMATICALLY RECORDING PULSE RATE

(including the cheek, back and abdomen) are recorded by means of thermo-couples in contact with the body, and connected to a potentiometer in an adjacent room by means of flexible wires, as shown in Fig. 1. The rectal and surface temperature readings were taken without the knowledge of the subjects. A permanent record was made of the pulse on smoked paper whenever desired, as illustrated in Fig. 2.

In addition to counting the rate of respiration by an observer, the rate was also recorded at intervals unknown to the subject by means of a kymographion. An inflated tube is strapped around the chest, and connected by means of long rubber tubing to the kymographion, which is placed in the room adjoining the chamber. The wave

lengths are traced on smoked paper encircling the drum of the instrument, as can readily be seen in Fig. 3.

In determining the basal metabolic rate, many difficulties were encountered, as it involved the transportation of the expired breath of the subject undergoing the test to the apparatus in an adjacent room. It was also necessary to collect for analysis a correct average sample of the expired air of the lungs. This was accomplished by having the subject breathe through the mouth-piece of a Gibbs oxygen breathing apparatus, which was connected by means of 1¼ in. rubber tubing passing through the wall of chamber to a graduated gasometer, Fig. 4. This mouth-piece consists of two butterfly valves, one of which opens when the subject inhales and closes when he exhales, while the other

FIG. 3. KYMOGRAPHION; RECORDING RATE OF RESPIRATION

closes when the subject inhales and opens when he exhales, thus conveying the exhaled air to the gasometer. A small nose-clip prevented nasal breathing during the collection of the samples. Three samples of 10 liter each were collected from each subject at one-hour intervals during the test. From each of these an average sample was then analyzed for CO_2 and O_2 on a Haldane apparatus[3].

Sweat was collected from the chest, arms and face by scraping the skin lightly with a glass beaker. No previous cleansing of the skin or hair was performed. After the collection, which usually lasted 15 min. at the beginning and at the end of the test, the fluid was poured into a test tube and was tightly stoppered.

The blood pressure was measured by means of a mercury sphygmomanometer, the auscultatory method with the subject in a sitting position being used.

All blood work, with the exception of making counts, was done by W. P. Yant, and a complete report of these findings will be the subject of a later paper.

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[3] Bureau of Mines Bul. 42. The Sampling and Examination of Mine Gases and National Gas by G. A. Burrell, and F. M. Seibert.

The amount of air motion in the room was measured by means of the Kata thermometer, shown in Fig. 5.

The temperature and humidty of the air in the test chamber was determined by means of an automatic indicating psychrometer, shown by Fig. 6.

Analysis of the air in the chamber was made at intervals during the experimental work by W. F. Holbrook under the supervision of G. W. Jones. It was believed that the presence of subjects in the

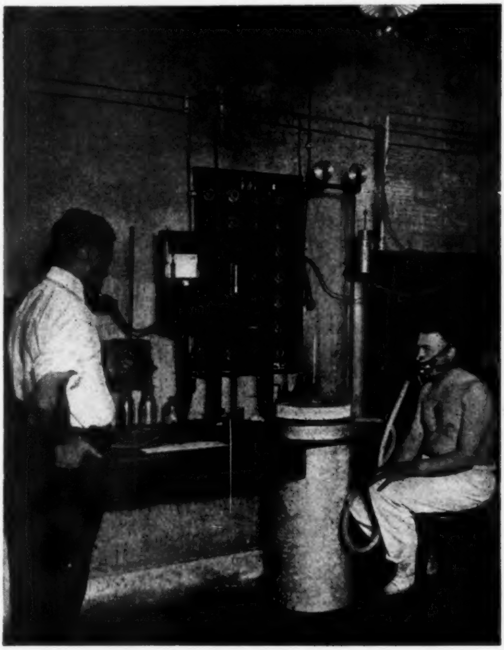


FIG. 4. METABOLISM APPARATUS FOR ANALYZING EXHALED AIR

chamber would perhaps reduce the oxygen and increase the carbon dioxide content of the atmosphere, so that symptoms complained of could be attributed to a low percentage of oxygen and a high percentage of carbon dioxide. This, however, did not occur. Samples of the air were taken by the standard Bureau of Mines vacuum-tube method, and analyzed on a modified Haldane gas apparatus in the same manner as analysis of mine air is made.

Analyses show that the maximum amount of carbon dioxide found, even at the end of the experiment, was never greater than 0.17 per cent, and that the oxygen never fell below 20.72 per cent, showing only very small changes from normal air.

PROCEDURE

Human subjects are used for all tests. Those who are exposed to the temperature change first rest for a period of 2 hr. in one of the chambers, where the atmospheric conditions maintained are similar to those in any well-ventilated living room. Here the pulse and the respiratory rate, body and surface temperatures, and blood pressure are recorded, in order to learn the conditions of each subject before the experiment. This preliminary *conditioning,* as it may be called, also has the tendency to eliminate body changes caused by previous activities and environment.

At the end of the 2 hr. period the subjects are accurately weighed before entering the test chamber. During the high temperature experiments the subjects strip to the waist, thus approximating the con-

FIG. 5. KATA THERMOMETER FOR MEASURING AIR MOTION

ditions of men working in high temperatures in the industries. The subjects remain in the test chamber for a period of 3 hr. whenever possible. In some experiments, where the temperature is high and is accompanied by a high humidity, the subjects are not able to remain therein for the entire period—in fact, death might result before the lapse of 3 hr. In these tests the subjects remain until the conditions become unbearable. Frequently in these severe tests the pulse rate exceeds 160 beats per min., and the body temperature reaches 104 deg. fahr. On leaving the chamber the subjects are again weighed, and a record is kept of the fall in body temperature and pulse rate. The subjects are shown in the test chamber in Fig. 7.

Time will not permit of a discussion of every experiment, but a few charts and graphs are reproduced and reveal pertinent facts of interest.

Only one dry and wet bulb reading with the relative humidity is given for each test. This is possible, because whenever these readings varied by more than ½ deg., the experiment was discarded. Ac-

curate record was kept of the readings, which were taken every few minutes, and the temperatures given are the average for each test.

DISCUSSION OF RESULTS

The first point that may attract attention is the great variations of external temperature which the human being can endure, and, at the same time, retain his physiological efficiency. The experiments indicated that the upper limit of man's ability to compensate for atmospheric conditions, when stripped to the waist and resting in still air, lies around 90 deg. fahr. saturated. The chart, Fig. 8, shows a beginning rise in body temperature and pulse rate at 90 deg. fahr. with relative humidity of 100 per cent.

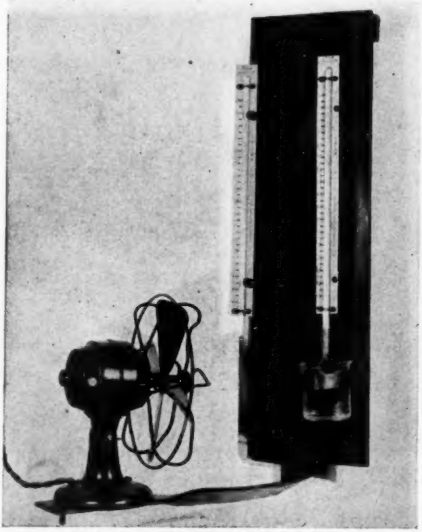


FIG. 6. AUTOMATIC PSYCHROMETER

Many recent published articles emphasize the significance of the wet bulb temperatures, and many investigators are inclined to overlook the dry bulb. Although the wet bulb is the more important of the two, still the dry bulb cannot be ignored. The relative importance between the dry and wet bulb temperatures has been sufficiently demonstrated by the equal comfort lines in the article by Houghten and Yagloglou, Determining Lines of Equal Comfort.

A comparison of Experiment 51 with 82 and 62 with 81 shows the physiological changes. For instance in Experiment 51 where the wet bulb was 90.4 deg. fahr. (dry 104.08 deg. fahr.) one subject's rectal

temperature rose to 100.80 deg. fahr. and his pulse to 126 pulsations per min. in 3 hrs., while in Experiment 82 where the wet bulb was 90.3 deg. fahr. (dry 90.3 deg. fahr.), the same subject's rectal temperature only rose to 99.2 deg. fahr. and his pulse to only 94 pulsations per min. in the same period of time. In another group, Experiment 62 where the wet bulb was 81.61 deg. fahr. (dry 110.0 deg. fahr.) the rectal temperature rose to 99.6 deg. fahr., pulse 90, while in Experiment 81 where the wet bulb was 81.8 deg. fahr. (dry 144.7) the rectal temperature rose to 100.4 deg. fahr., and the pulse to 108.

The length of exposure to different high temperatures is a determining factor in the severity of the after depression produced in an

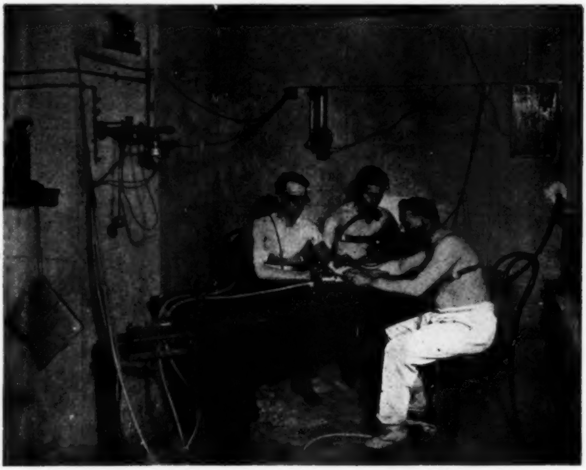


FIG. 7. SUBJECTS IN HOT TEST CHAMBER

individual. Although the physiological reactions may deviate further from the normal in higher temperatures for a short period of time than in a temperature not so high, for a longer period, the exhaustion resulting from the latter test is more prolonged than in the former.

In Experiment 47, where the subjects remained 1½ hr., and the rectal temperature rose to 101.4 deg. fahr. and the pulse to 149 pulsations per min., the fatigue and weakness which followed lasted for a longer period than the fatigue and weakness which followed in Experiment 63 where the rectal temperature rose to 103 deg. fahr. and the pulse to 162, but for a period of 45 min.

Of the physiological disturbances that occur when subjects are exposed to abnormal atmospheric conditions, the results of the experiments emphasize the importance of the circulatory system. Were this work confined to a single physiological measure of discomfort due to high temperature—in the light of present experimental evidence—the pulse rate would be selected as probably the best index.

Investigators have long recognized that the body temperature alone is not the cause of the discomfort, and their conclusions as a whole correspond with those of Haldane[4], who stated that the discomforts produced by high temperatures undoubtedly depend to some extent on other causes than the rise of body temperature, as indicated by the rectal temperature.

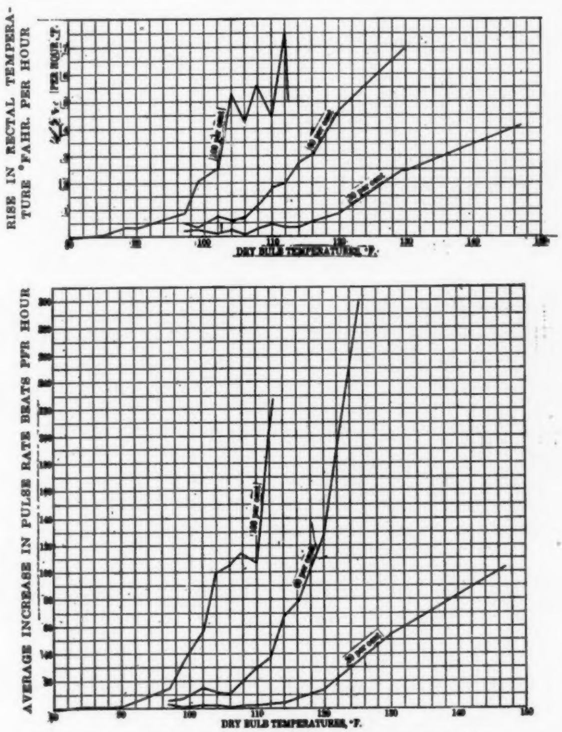


FIG. 8. CHARTS SHOWING CHANGES IN BODY TEMPERATURE AND PULSE RATE

It frequently so happened in these experiments that the rectal temperature rose slightly after leaving the chamber, but the subject nevertheless, felt more comfortable with the fall in pulse rate. Without exception, every subject began to complain of unbearable symptoms when the pulse-rate materially increased, irrespective of the rise in body temperature. Though no arbitrary rate at which certain symptoms occur has been determined, the consensus of opinion—as expressed by the subjects themselves—intimates that very uncomfortable sensations are felt after the pulse-rate exceeds 135 pulsations per min., and the atmospheric condition becomes unbearable when the rate exceeds 160 per min.

Reference to the results obtained in Experiments 77, 78 and 80 reveal the fact that some subjects attain a high pulse rate much sooner than others and therefore are compelled to leave the chamber sooner. In one instance one man attained a pulse rate of 160 in 10

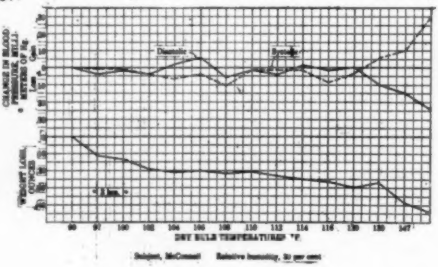


FIG. 9

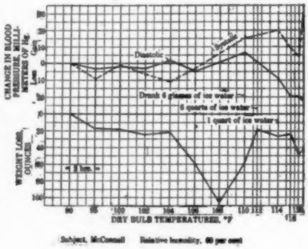


FIG. 10

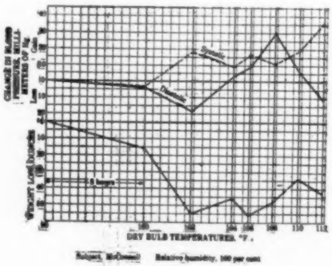


FIG. 11

CHARTS SHOWING COMPARATIVE LOSS IN WEIGHT AND CHANGES IN BLOOD PRESSURE FOR DIFFERENT TESTS

min., while another remained in the chamber 35 min. before the pulse reached 180. In the next experiment one attained a pulse rate of 160 per min. in 55 min. while another remained 1½ hr. and the pulse only reached 156 pulsations per min. In Experiment 60 the pulse of one reached 142 in 25 min. while the pulse of another reached 156 in 45 min. The pulse-rate rapidly diminished on leaving the test chamber.

The rise in blood pressure no doubt contributes to the discomfort. Figs. 9, 10 and 11 best show the effect of different temperatures on the blood pressure. It is seen by these tracings how the systolic pressure rapidly rises with the increase in temperature, while the diastolic pressure falls. At 30 per cent relative humidity the change begins around 120 deg. fahr., at 60 per cent around 106 deg. fahr., and at 100 per cent around 100 deg. fahr. Subjects describe the sensation felt on reaching the unbearable condition as follows: First, a slight palpitation of the heart occurs, which increases in severity until a feeling of *floating in the air* is experienced. This is accompanied by dizziness, and frequently with a numbness or soreness of the face, and with nausea.

J. S. Haldane, Abs. from *The Journal of Hygiene*, Vol. V, 1905, pp. 494-513.

The body temperature is apparently not an influencing factor in these instances. It will be observed that in Test 80, Fig. 12, the subjects were able to tolerate the condition only a comparatively short time, although the body temperature was not so high as in Test 63, Fig. 13. In the first the rectal temperature only rose 2.5 deg. fahr. whereas in the second the rectal temperature rose almost 4.5 deg. fahr. The pulse rate, on the other hand, increased approximately 80 beats in each case. In these experiments the subjects remained as long as they could endure the discomfort.

The charts of each experiment and the composite charts demonstrate the parallelism between the atmospheric temperature and the body temperature perhaps better than any explanation. It will be seen that neither the mouth nor rectal temperatures reached any constant value, but continued to rise as long as the subject remained in the chamber. In some tests, usually those with a low humidity, the rectal temperature continued to rise a little after leaving the chamber. This is perhaps due to the fact that the subjects changed from a resting stage to an active one. The rectal temperature after the slight rise, presently began to drop gradually to normal; while on the other hand the mouth temperature dropped almost immediately to normal. Mouth temperatures are very unreliable and should not be used for scientific work, except for comparison only. Tests 80 and 59, as shown in Figs. 12 and 14, illustrate the small rise and gradual fall of the rectal temperature and the sudden drop of the oral temperature. No graphs of the changes in surface temperatures are included in this paper. Sufficient data have not been collected to justify any conclusions, and the subject of surface temperatures, after more work has been done, will be submitted in a separate article. In general, it may be stated that the surface temperature rises when the subject enters the hot chamber, but falls again as soon as he begins to perspire, and again rises slowly.

The changes in the respiratory rate in the subjects do not exactly parallel with those of other investigations. No marked increase in respirations were found during the test. A new subject, however, when asked to record his respirations, would invariably give a high rate, which did not correspond to the rate obtained by the kymographion, or when the subject's respirations were counted unknown to him. In one instance the subject counted 44 respirations per min., when normally he was breathing at the rate of 12. On the other hand, on leaving the test chamber, and entering into a cooler room, the respirations increased for a short period, but in a short time returned to normal.

The loss in weight is represented on the charts by the difference, measured in ounces, between the weight on entering and on leaving the chamber. It is interesting to note that the loss of weight varied with the individual, the heavier and stouter man losing more than the light and thin one. Notwithstanding this, the lighter man as a rule could not endure the temperature conditions as long, and complained more of the exhaustion which follows. Figs. 9, 10 and 11

show the comparative loss in weight for the different tests. It will be observed that the loss gradually increased with an increase in atmospheric temperature. Whenever the subject drank ice water he immediately gained in weight. The subject usually regained the entire weight lost within 24 hr.

A table prepared by Mr. Holbrook on basal metabolism is given here merely to show the extent of the work done so far. The results are not conclusive, and further work will necessarily have to be carried on before any relationship between the atmospheric and metabolic processes within the body, if such exists, will be established. During this work the subjects had always eaten at least 3 hr. previous to taking the first sample. The results are presented in the following table:

GAS VOLUMES CALCULATED DRY AND AT N. P. T.

Rate of Respiration	Volume per Breath, cc.	Alveolar Air, mm.	Oxygen consumed per Min. cc.	Respiratory Quotient	Dead Air Space	Calories per hour	Time Elapsed hours
Experiment No. 35			Date—6-29-22			T. 76°F. R.H. 72%	
1	2	3	4	5	6	7	8
			McConnell				
11	639	......	242	0.928	......	72.0	0.33
11	502	......	184	0 967	......	55.4	1.40
Experiment No. 41			Date—7-18-22			T. 97°F. R.H. 63%	
1	2	3	4	5	6	7	8
			McConnell				
10	459	......	179	0.870	......	52.5	0.55
10	768	......	266	0.980	......	80.2	1.80
12	726	......	290	0.975	......	87.2	2.75
			Fulton				
11	524	......	298	0 768	......	85.2	0.67
11	613	......	327	0.819	......	94.6	1.95
12	473	......	277	0.792	......	79.6	2.92
Experiment No. 42			Date—7-20-22			T. 97°F. R.H. 40%	
1	2	3	4	5	6	7	8
			McConnell				
12	566	......	246	0.883	......	72.4	0.08
11	562	......	235	0.856	......	68.7	1.85
10	520	......	201	0.844	......	58.6	2.73
			Flinn				
15	476	......	279	0.820	......	80.8	0.37
13	479	......	275	0.792	......	79.1	2 00
15	519	......	307	0.758	......	87.5	2.95
Experiment No. 43			Date—6-21-22			T. 97°F. R.H. 35%	
1	2	3	4	5	6	7	8
			McConnell				
13	632	......	260	0.942	......	77.6	0 08
11	550	......	217	0.922	......	64.6	1.45
11	763	......	221	1.140	......	69.2	2.77
			Fulton				
10.5	500	......	225	0.782	......	64.5	0.20
12	461	......	203	0.812	......	58.7	1.62
13	495	......	230	0.805	......	66.4	2.90

Rate of Respiration	Volume per Breath, cc.	Alveolar Air, mm.	Oxygen consumed per Min. cc.	Respiratory Quotient	Dead Air Space	Calories per hour	Time Elapsed hours
Experiment No. 45			Date—8-10-22			T. 90°F. R.H. 60%	
1	2	3	4	5	6	7	8
			McConnell				
11.5	471	......	199	0.863	......	58.2	0.12
13	610	......	273	0 926	......	81.2	1.67
13	625	......	256	0.964	......	76.8	2.42
			Fulton				
12.5	477	......	271	0.737	......	77.0	0.28
15	606	......	344	0.901	......	101.6	1.83
13	559	......	305	0.835	......	88.6	2.58
Experiment No. 46			Date—8-11-22			T. 88°F. R.H. 95%	
1	2	3	4	5	6	7	8
			McConnell				
11.5	596	......	290	0.748	......	82.4	0.42
12.5	492	......	234	0.865	......	68.6	1.83
12	532	......	237	0.889	......	69.8	2.75
			Fulton				
13	482	......	288	0.823	......	83.4	0.55
12.5	528	......	262	0.885	......	77.1	2.05
12.5	510	......	279	0.793	......	80.2	2.92
Experiment No. 47			Date—8-15-22			T. 99°F. R.H. 99%	
1	2	3	4	5	6	7	8
			McConnell				
10	602	34.3	254	0.850	93	74.2	0.0
10	843	26.1	168	1.618	23	......	1.03
			Fulton				
13	466	34.5	287	0.777	61	82.0	0.0
12.5	666	38.5	358	0.850	156	99.6	0.97
Experiment No. 48			Date—8-17-22			T. 99°F. R.H. 32.7%	
1	2	3	4	5	6	7	8
			McConnell				
11.5	523	26.7	225	0.876	7	66.1	0.0
10	623	28.2	251	0.856	4	73.4	1.67
11.5	613	......	327	0.683	......	91.6	2.50
			Fulton				
12.5	466	38.5	281	0.762	104	80.0	0.0
15.5	630	39.5	354	0.922	195	105.2	1.67
15.5	601	......	372	0.790	......	106.9	2.50
Experiment No. 49			Date—8-18-22			T. 101.9° F. R.H. 33.1%	
1	2	3	4	5	6	7	8
			McConnell				
10.5	509	29.2	223	0.814	12	64.5	0.0
13	521	30.1	259	0.832	56	75.2	1.50
9.5	618	......	228	0.838	......	66.3	2.47
			Fulton				
12.6	555	39.5	340	0.765	203	97.1	0.0
13	521	40.4	280	0.836	138	81.4	1.37
14.5	450	......	239	0.840	......	69.6	2.22

Rate of Respiration	Volume per Breath, cc.	Alveolar Air, mm.	Oxygen consumed per Min. cc.	Respiratory Quotient	Dead Air Space	Calories per hour	Time Elapsed hours
Experiment No. 53			Date—8-29-22		T. 102°F. R.H. 99.2%		
1	2	3	4	5	6	7	8
			McConnell				
12	643	307	305	0.812	88	88.1	0.0
14	1242	265	409	1.118	193	126.4	1.0
			Fulton				
13.5	517	404	311	0.826	125	90.1	0.0
9	857	352	346	0.806	136	99.8	0.92
Experiment No. 54			Date—8-31-22		T. 104°F. R.H. 30%		
1	2	3	4	5	6	7	8
			McConnell				
12.5	602	324	305	0.830	97	88.5	0.0
11.5	462	283	227	0.840	−12	66.0	0.97
11.5	549	301	254	0.824	62	73.6	2.02
12	520	290	259	0.799	39	74.6	2.87
			Fulton				
13	618	403	344	0.852	166	100.5	0.0
11.5	545	361	304	0.764	91	86.8	1.00
15	502	397	317	0.735	180	89.8	2.05
14.5	619	361	381	0.822	136	110.3	2.92
Experiment No. 55			Date—9-1-22		T. 104°F. R.H. 100%		
1	2	3	4	5	6	7	8
			McConnell				
12	557	36.9	292	0.785	139	83.8	0.0
15	1257	21.1	520	1.020	−64	159.7	0.75
			Fulton				
13.5	524	38.9	322	0.821	119	93.2	0.0
32	612	34.5	602	0.899	214	177.8	0.75
Experiment No. 57			Date—1-14-22		T. 106°F. R.H. 30%		
1	2	3	4	5	6	7	8
			McConnell				
11	651	33.6	303	0.810	114	87.5	0.0
12.5	542	29.0	274	0.784	60	78.6	1.03
10.5	583	34.7	266	0.786	117	76.4	2.03
14	465	29.0	257	0.817	50	74.4	2.85
			Fulton				
13	910	35.9	500	0.860	174	144.8	0.0
14	465	42.1	258	0.705	207	72.6	1.02
15.5	450	34.7	289	0.776	108	82.8	2.02
12.5	830	38.9	398	0.933	215	118.5	2.83
Experiment No. 59			Date—9-21-22		T. 108°F. R.H. 29.9%		
1	2	3	4	5	6	7	8
			McConnell				
11.5	619	34.0	341	0.759	87	97.2	0.0
11.5	520	32.4	273	0.695	108	76.7	0.98
11	562	32.7	284	0.695	119	79.0	2.13
12	694	30.9	335	0.683	196	94.0	3.00

Rate of Respiration	Volume per Breath, cc.	Alveolar Air, mm.	Oxygen consumed per Min. cc.	Respiratory Quotient	Dead Air Space	Calories per hour	Time Elapsed hours
			Fulton				
13	640	378	424	0.774	104	121.3	0.0
13.5	560	352	332	0.681	141	93.0	1.00
15.5	591	405	393	0.789	189	112.9	2.15
13	572	386	359	0.702	164	100.9	2.98
Experiment No. 60			Date—9-22-22			T. 108°F. Sat'd.	
1	2	3	4	5	6	7	8
			McConnell				
11.5	566	33.7	327	0.704	89	91.8	0.0
9	814	30.7	372	0.654	117	103.5	0.53
			Fulton				
15.5	689	41.5	484	0.820	195	140.2	0.0
15	719	37.9	503	0.782	169	144.4	0.50
Experiment No. 58			Date—1-15-22			T. 106°F. R.H. 60%	
1	2	3	4	5	6	7	8
			McConnell				
12.5	628	34.2	335	0.764	144	95.6	0.0
13	628	30.1	340	0.726	107	96.1	1.20
12.5	646	29.9	338	0.727	114	95.6	2.05
14	874	27.2	403	0.836	163	117.2	......
Experiment No. 50			Date—8-22-22			T. 102°F. R.H. 60%.	
1	2	3	4	5	6	7	8
			McConnell				
10.5	573	27.7	279	0.834	-66	81.1	0.0
9.5	706	30.2	306	0.811	17	88.4	1.0
9.5	705	......	290	0 817	......	84.2	2.0
11.0	637	26.0	307	0.766	-21	87.6	3.0
Experiment No. 51			Date—8-24-22			T. 104.8°F. R.H. 60%	
1	2	3	4	5	6	7	8
			McConnell				
11.0	528	32.9	257	0.797	51	74.0	0.0
9.5	720	30.7	288	0.810	55	83.2	0.80
7.0	880	30.9	280	0 788	28	80.4	1.80
7.5	905	31.1	292	0.812	56	84.4	2.82
Experiment No. 52			Date—8-25-22			T. 102°F. R.H. 37%	
1	2	3	4	5	6	7	8
			McConnell				
13	636	29.7	277	0.890	96	81.6	0.0
11	726	31.9	328	0.817	87	94.9	0.92
9	659	29 6	234	0.785	72	67.2	1.92
11	608	31.2	305	0.820	91	88.3	2.75

SWEAT DETERMINATIONS

Measurements of volume, acidity, and chloride concentration were made within a few hours of the collection of the sample. The acidity was estimated very roughly calorimetrically, using phenolsulphonephthalein as the indicator. Chlorides were titrated by the Harvey

modification of Volhard's method[5]. These determinations were made by Dr. E. F. Adolph, of the University of Pittsburgh, and his conclusions follow the data. The data are presented in the accompanying table of Sweat Analysis:

The experiments indicate the following general results:

1. With regard to the volume of sweat secreted:
   a. The rate of secretion mounts rapidly with increased wet bulb temperature.
   b. Sweat can be collected in larger amounts from the chest than from the face under all conditions. The sweat from the arms is almost as copious as from the chest.
2. With regard to the actual acidity of the sweat:
   a. The sweat becomes more alkaline as its secretion continues.
   b. The secretions from the chest and arms are more acid than from the face.
   c. The range of acidity found does not depart greatly from the neutrality. It is more often on the alkaline side.
3. With regard to the chloride concentration of sweat:
   a. The sweat becomes more concentrated with time.
   b. The most dilute sweat occurs in saturated atmospheres. This is probably accounted for by the evaporation during collection. The most dilute sweat contained 0.044 m. chloride, equivalent to 0.26 per cent NaCl.
   c. The salinity and alkalinity are roughly proportional to each other.
   d. Sweat from the chest is often of higher concentration than that from the arms.

SWEAT ANALYSIS

FULTON

				1st 15 min.			2nd 15 min.		
Test No.	Humidity %	W. B.T. of	Place	Vol.	pH.	cl. con.	Vol.	pH.	cl. con.
68	30	85					10		122
			Chest	4	6.5		3	7.8	
78	30	96	Arms	8	6.5	57	4	7.8	
			Face	2	6.7		2	7.8	
76	30	97	Arms		7.0	83		7.7	
			Chest	10	6.5	86	10	7.5	103
75	30	108	Arms	10	6.5	68	10	7.5	93
			Face	7	7.5	104	8	7.5	136
70	60	99					40	7.7	87
72	60	105		20	7.9	55	20	7.9	77
			Chest	6		71	4		
77	60	113	Arms	5		71	5		
			Face	5			3		
69	100	113		20		44	20		66

[5] *Archives of Internal Medicine*, Vol. 6, p. 12, 1910.

McCONNELL

Test No.	Humidity %	W. B.T. of	Place	1st 15 min. Vol.	pH.	cl. con.	2nd 15 min. Vol.	pH.	cl. con.
68	30	85					4		
			Chest	6	6.8	69	6	7.6	107
78	30	96	Arms	6	6.8	68	3	7.6	
			Face	5	7.6	69	4	8.0	
76	30	97	Arms		7.5	82		7.9	
			Chest	10	6.5	77	10	7.5	101
75	30	108	Arms	10	6.5	70	10	7.5	108
			Face	6	7.5	78	7	7.5	105
70	60	99					44	7.4	99
72	60	105		20	7.0	54	8	7.5	79
			Chest	7		62	3		
77	60	113	Arms	8		55	4		
			Face	5		67	3		
69	100	113							

SMITH

Test No.	Humidity %	W. B.T. of	Place	1st 15 min. Vol.	pH.	cl. con.	2nd 15 min. Vol.	pH.	cl. con.
68	30	85					2		
			Chest	6	7.5	102	2	7.8	
78	30	96	Arms	6	7.8	93	1	7.8	
			Face	3	7.8		2	7.8	
76	30	97	Arms		7.6			7.5	
			Chest	5	7.5		7	7.5	142
75	30	108	Arms	3	7.5		8	7.5	136
			Face	2	7.5		3	7.5	
70	60	99					12	7.9	124
72	60	105		15	7.7	80			
			Chest	3					
77	60	113	Arms	2					
			Face	3					
69	100	113					13		68

According to Hunt[6], the sweat as it leaves the glands is never more concentrated in chloride than 0.20 per cent NaCl. Higher concentration he finds are due to evaporation during collection and to residual solutes clinging to the skin. The acidity of sweat has been studied extensively by Kittsteiner[7], and Talbert[8]. Both find a definite regional variation, which agrees with the present results. Talbert, however, claims that the sweat is always more acid than the blood. This does not seem to be true, even where the escape of volatile acids from the sweat is prevented.

The question of chief interest is whether the sweat glands secrete a product which is constant under all conditions or whether they regulate carefully the fluid composition of the body as the kidneys do. The first conception is not proven wrong by any of the data at hand, for the variations in acidity and salinity may be due to chemical changes in the blood equilibria at the corresponding times. The sweat is always more dilute than the blood and it is probable that under

[6] *Journal of Hygiene*, Vol. 12, p. 479, 1912.
[7] *Archives of Hygiene*, Vol. 73, p. 275, 1911.
[8] *Amer. Journal of Physiology*, Vol. 59, p. 454, 1922.

proper conditions of collection the sweat always has the same concentration of each constituent, as Hunt concludes.

There is no correlation between composition of sweat and skin temperature, though the temperature rises during the continuance of sweating. It is to be concluded, therefore, that only the rate of sweating varies with the atmospheric conditions. The loss of substances

CHARTS SHOWING ACTION OF RECTAL AND ORAL TEMPERATURES AND RISE IN PULSE RATES BEFORE, DURING, AND AFTER TESTS

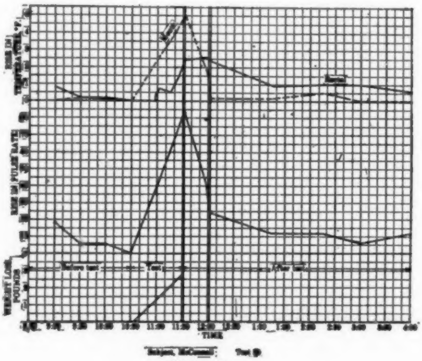


FIG. 12

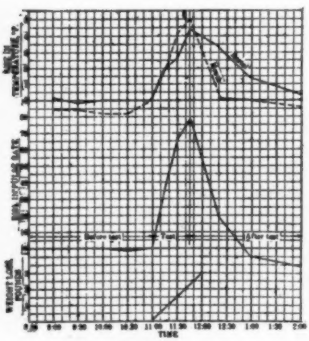


FIG. 13

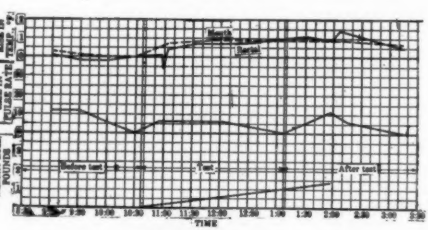


FIG. 14

from the body through the sweat cannot be changed in their proportions to each other.

Blood was taken from the subjects' fingers, at half hour intervals, and the sugar and haemoglobin content was determined. The haemoglobin was estimated by the spectrophotometric method, and it was found that the haemoglobin increased proportionately with the amount of weight lost.

A modification of Maclean's method was used to determine the sugar content of the blood. The results obtained, however, had no significance for the tests in high temperatures. A complete report of this work, as stated before, will be the subject of another paper.

Red and white blood, and some differential leucocyte counts, were made on one subject before and after some of the tests. A relative increase of all cells was found, but not an absolute one. The greatest

increase was found after the greatest loss in weight. The counts soon returned to normal after the subject satisfied his thirst for water. This phase of the work will be continued, as hardly sufficient work has been done to justify any conclusions.

Samples of urine passed after some of the severe tests were analyzed for albumen and sugar. Frequently during the experiments the subjects complained of an intense desire to urinate. The urine passed, however, was scanty and high colored. The specific gravity in some instances reached as high as 1.030. The samples examined were all negative for albumen and sugar.

The chief symptoms complained of under the adverse conditions of temperature were as follows: The subjects who, frequently, on entering the chamber, were in a happy mood and of a joking disposition, soon became restless and irritable. They complained of a headache and palpitation of the heart. The headache soon became throbbing in nature, and the palpitation distressing. Great thirst was experienced. The eyes became inflamed and sore. A feeling as of a weight on the chest was noticed. The voice suffered somewhat in that it became an effort to speak. Dizziness and confusion followed. After leaving the chamber it was necessary to sit down and rest for 5 or 10 min. before taking a shower. Weakness and a *dragged out* feeling continued for some time, depending upon the severity of the test. A metallic taste was a noticeable symptom, and persisted for 1 or 2 hr. following the high temperature experiments.

While irrelevant to the present study, an attempt was made to determine the cause of the inflammation of the eyes, and also to find out whether or not the drinking of ice water produced cramps, as so frequently complained of by workers in the industries.

On a day when the eyes were normal, a small quantity of sweat which had been collected during an experiment was dropped in the eye. The conjunctiva almost immediately became inflamed and the eye felt sore. The sweat collected from the face seemed less irritable to the eye than that collected either from the chest or arms. Sweatbands were then applied to the forehead during the experiments, these preventing the sweat from falling into the eyes and also prevented the inflammation. This proves sufficiently that the sweat is the chief factor in producing the conjunctivitis which accompanies exposure to heat.

It has frequently been stated that workers exposed to high temperatures developed severe cramps after drinking ice water. A few of the subjects of these experiments volunteered to drink ice water after about an hour's exposure to high temperature, and two of them. in one experiment, drank a quart of ice water in less than 15 min. without ill effects. Cramps did not develop in any of the subjects at any time. In reference to the experiments which give the results of drinking ice water, the results show that, while the water satisfied the subjects' thirst, the pulse rate and rectal temperature continued to rise. The water apparently had no effect in delaying these.

SUMMARY

1. There is an inability of the body at rest and in still air to compensate for saturated atmospheric conditions exceeding 90 deg. fahr.

2. The physiological effects resulting from exposure to high temperatures and humidities depend upon both the wet and dry bulb temperature readings.

3. The exhaustion and weakness following subjection of human beings to a very high temperature and humidity for a short period is not so severe as subjection to a moderately high temperature and humidity, but for a longer period.

4. The highest dry and wet bulb temperatures attained and length of time endured in the experiments are as follows:

Dry Bulb	Wet Bulb	Relative Humidity	Time, Minutes
112.5	112.5	100	35
120.2	104.02	60	40
147	108.4	30	45
157	100.43	15	45

5. The pulse-rate, rather than the rise in body temperature, apparently determines the extent of the discomfort experienced by the subject. Subjects became very uncomfortable after the pulse-rate exceeded 135 pulsations per min., and complained of unbearable and distressing symptoms when the pulse exceeded 160 per min. The highest pulse rate recorded was 184 per min.

6. The systolic and diastolic blood pressure fell with moderate increase in temperature and humidity, and the systolic rose and the diastolic fell, thus increasing the pulse pressure, in high temperatures and humidities.

7. The curve of rectal temperature shows a close parallelism with the curve of the temperature of the test chamber at any constant humidity.

8. Subjection to high temperatures and humidities produced no marked change in the respiratory rate.

9. Loss of weight varies with the individual, the loss increasing with the length of exposure to high temperatures and with the severity of the test.

10. The haemoglobin content of the blood increases proportionately to the loss in weight.

11. An estimation of the changes in the sugar content of the blood proved negligible.

12. A relative increase in the blood count was found, but not an absolute one.

13. The rate of sweating varies with the atmospheric conditions. The loss of substances from the body through the sweat do not change in their proportions to one another.

14. Examination of the urine was negative for albumen and sugar, and specific gravity increased.

15. The sweat is the chief factor in producing the irritation and inflammation of the eyes.

16. Ice water, even when large quantities were swallowed, did not cause cramps under the conditions of the experiments.

ACKNOWLEDGMENTS

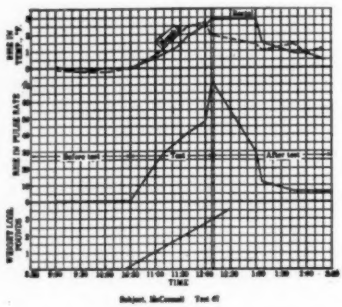
The valuable assistance and constructive criticism of a number of interested scientists have largely contributed to the success of this work. To Dr. L. R. Thompson, officer in charge of the Department of Industrial Hygiene and Sanitation, and to Dr. R. R. Sayres, chief surgeon, U. S. Bureau of Mines, under whose supervision the physiological studies are conducted, and whose assistance and encouragement were never lacking, we are gratefully indebted. Acknowledgment is also made to F. Paul Anderson, dean of the College of Engineering at the University of Kentucky, and director of the Research Laboratory of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS; and to Arno C. Fieldner, superintendent and supervising chemist of the Pittsburgh Experiment Station of the Bureau of Mines, for counsel and suggestions. Dr. William H. Howell of Johns Hopkins University, Dr. Ellsworth Huntington of Yale University, Dr. E. F. Adolph of the University of Pittsburgh, and O. W. Armspach, formerly of the Research Laboratory, have likewise contributed many helpful suggestions.

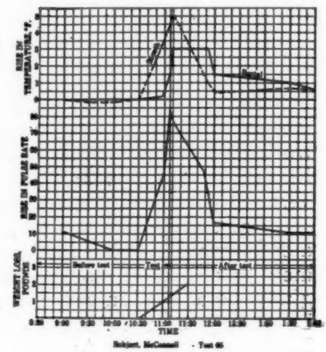
It is obvious that the results of these experiments would be misleading, and indeed even worthless, were not the men who maintain the temperature conditions, and those who act as subjects for the experiments, conscientious and accurate in their work. We wish to express our appreciation to C. P. Yagloglou, of the Research Laboratory, for the able manner in which he maintained the temperature conditions for each test, and for his accuracy in recording the physical data; to G W. Jones, W. P. Yant, W. F. Holbrook, L. B. Berger and W. B. Fulton, gas chemists of the Bureau of Mines, who were responsible for analyzing the air of the chamber, also for the basal metabolism and blood work on the subjects; to R. L. Lincoln and Herman Berg of the Research Laboratory, for assistance in maintaining the apparatus in working conditions; and to Dr. Adolph, of the University of Pittsburgh; and Drs. Sayers, Flinn, Smith, and Mitchell of the U. S. Public Health Service and Mr. Fulton of the Bureau of Mines, for frequently exposing themselves to the temperature conditions.

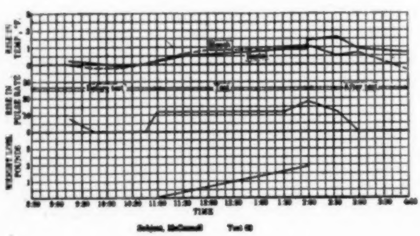
APPENDIX

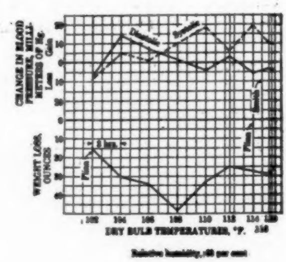
The original data taken during 41 tests is presented in the curves and tables grouped in this appendix. The rise in rectal temperature above the subject's normal, and the increase in pulse rate above normal, as observed during the tests, are plotted against time for each of the three subjects.

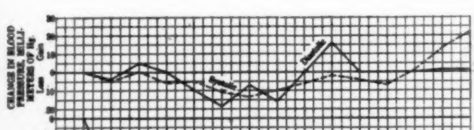
The small tables in several of the charts give the atmospheric conditions for the test and the normal temperature, the normal pulse rate, and loss in weight by perspiration for each subject.

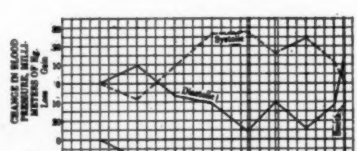




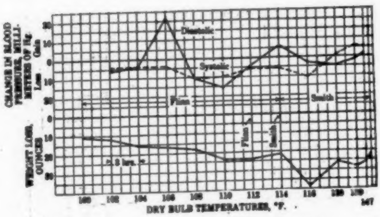




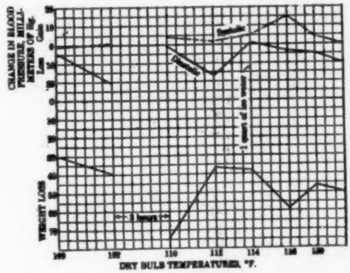




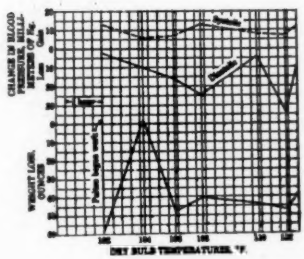
The rise in rectal temperature, increase in pulse rate, and loss in weight by perspiration for each subject during the period over which the observations were made are given in Table 1. The hourly rise in temperature, the hourly loss in weight by perspiration and the hourly increase in pulse rate in beats per minute are also given for the different subjects.



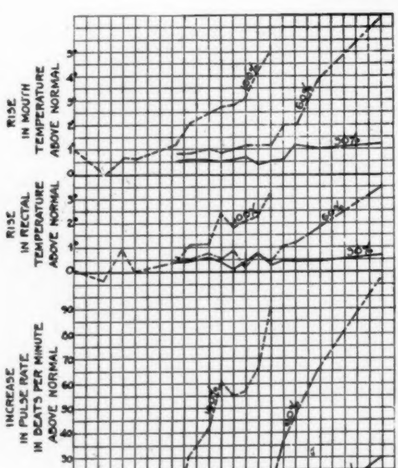
Relative humidity, 30 per cent



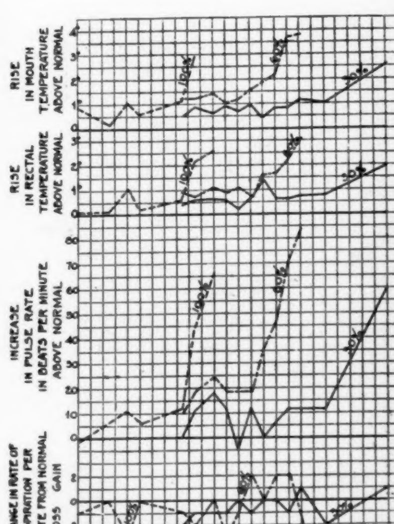
Subject, Fulton Relative humidity, 60 per cent



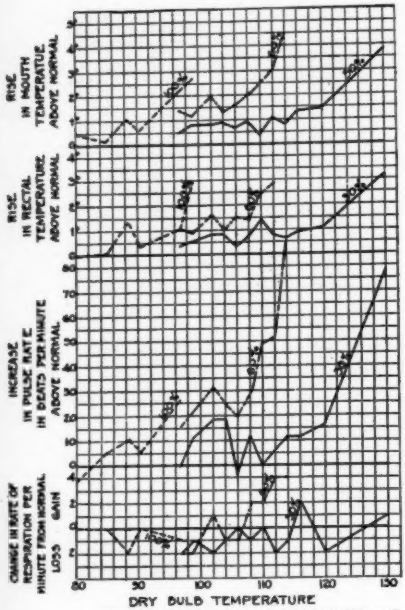
Subject, Fulton Relative humidity, 100 per cent



First half hour Subject McConnell

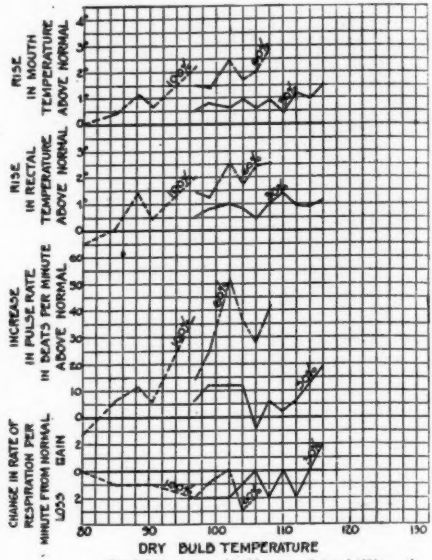


First hour Subject McConnell

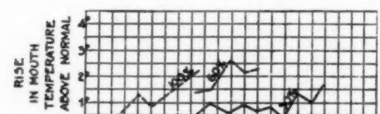


First One and one half hours Subject McConnell

First two hours Subject McConnell



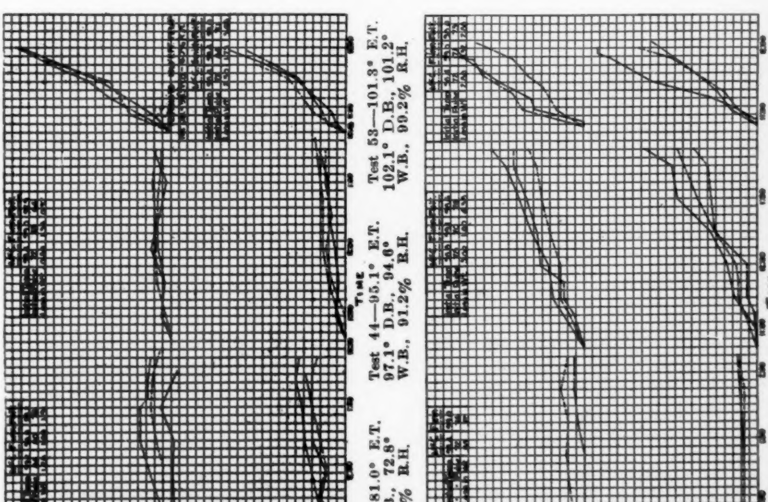
First two and one half hours Subject McConnell



First three hours Subject McConnell

Test 41—88.5° E.T. 96.9° D.B., 85.7° W.B., 63%

Test 42—83.2° E.T. 97.2° D.B., 77.0° W.B., 40% R.H.



Test 43—81.0° E.T. 96.9° D.B., 72.8° W.B., 31% R.H.

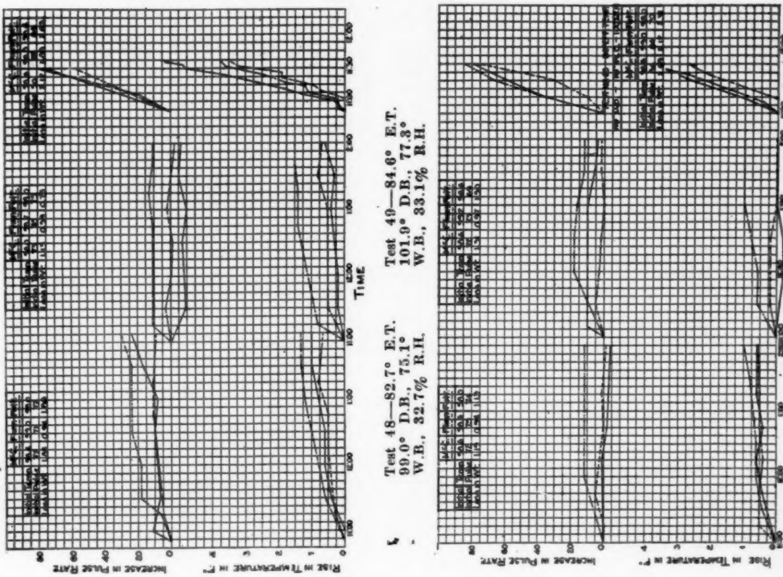
Test 44—95.1° E.T. 97.1° D.B., 94.6° W.B., 91.2% R.H.

Test 53—101.3° E.T. 102.1° D.B., 101.2° W.B., 99.2% R.H.

Test 45—89.3° E.T. 99.0° D.B., 86.2° W.B., 60% R.H.

Test 46—87.4° E.T. 88.3° D.B., 87.0° W.B., 95% R.H.

Test 60—107.8° E.T. 107.8° D.B., 107.7° W.B., 100% R.H.



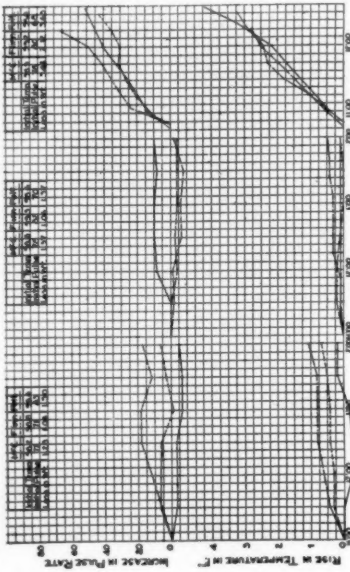
Test 48—82.7° E.T. 99.0° D.B., 75.1° W.B., 32.7% R.H.

Test 49—84.6° E.T. 101.9° D.B., 77.3° W.B., 33.1% R.H.

Test 54—85.2° E.T.
104.0° D.B., 77.3°
W.B., 30% R.H.

Test 57—86.5° E.T.
106.1° D.B., 78.8°
W.B., 30% R.H.

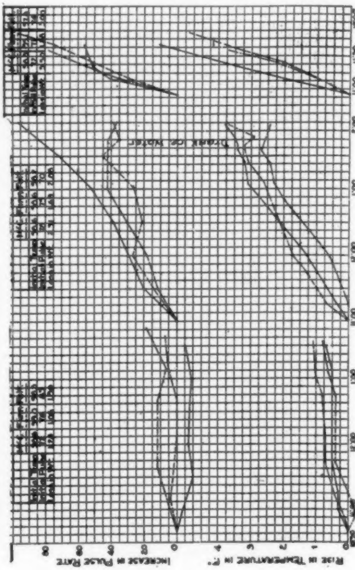
Test 47—99.0° E.T.
99.0° D.B., 98.0°
W.B., 99.9% R.H.



Test 59—87.5° E.T.
108° D.B., 79.8°
W.B., 29.9% R.H.

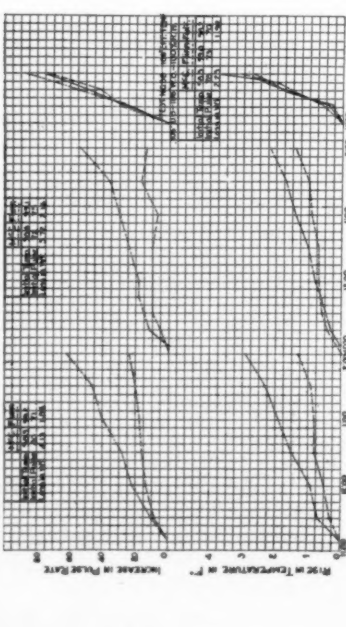
W.B., 60% R.H.
108.4° D.B., 94.5°
Test 61—96.0° E.T.

Test 55—104° E.T.
104° D.B. 104°
W.B., 100% R.H.



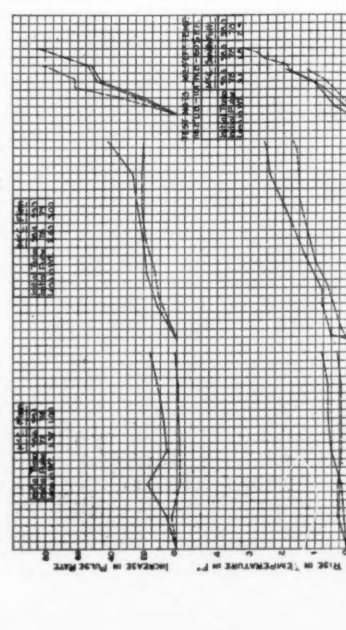
Test 50—91.0° E.T.
102.0° D.B., 89.0°
W.B., 60.1% R.H.

Test 51—93.2° E.T.
104.1° D.B., 90.4°
W.B., 60% R.H.



Test 52—85.3° E.T.
101.9° D.B., 78.5°
W.B., 37% R.H.

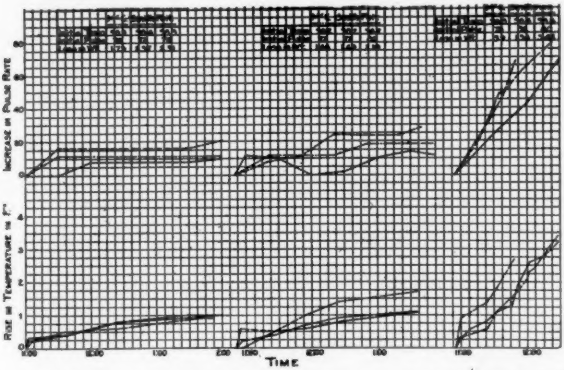
Test 58—94.9° E.T.
106° D.B., 92.2°
W.B., 60% R.H.



Test 62—88.9° E.T. 110° D.B., 81.6° W.B., 30% R.H.

Test 67—90.3° E.T. 112° D.B., 83.2° W.B., 30% R.H.

Test 76—101.8° E.T. 129.3° D. B. 97.5° W.B., 30% R.H.

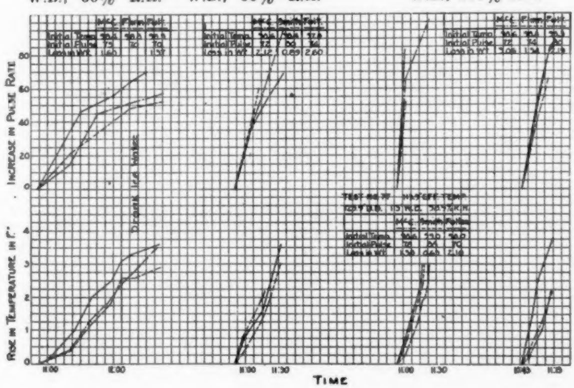


Test 66—99° E.T. 112° D.B., 96.7° W.B., 60% R.H.

Test 72—106.2° E.T. 120° D.B., 104.6° W.B., 60% R.H.

Test 65—112.0° E.T.

W.B., 100% R.H.



Test 68—91.4° E.T. 114.1° D.B., 84.6° W.B., 30% R.H.

Test 71—92.8° E.T. 116° D.B., 86.2° W.B., 30% R.H.

Test 78—101.1° E.T. 130° D.B., 96.4° W.B., 30% R.H.

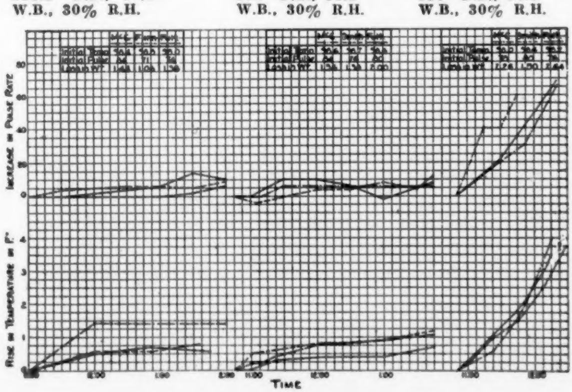


TABLE 1. SUMMARY OF RESULTS OF TESTS OF PHYSIOLOGICAL REACTIONS TO HIGH TEMPERATURES AND HUMIDITIES

Test No. Date 1922	Test Conditions	Subjects	TIME: Entered Chamber	TIME: Left Chamber	TIME: In Chamber (hrs.)	RECTAL TEMPERATURE: Normal	RECTAL TEMPERATURE: Total Rise	RECTAL TEMPERATURE: Time	RECTAL TEMPERATURE: Rate of Rise deg. per hr.	PULSE RATE IN BEATS PER MIN.: Normal	PULSE RATE IN BEATS PER MIN.: Total Increase	PULSE RATE IN BEATS PER MIN.: Time	PULSE RATE IN BEATS PER MIN.: Rate of Increase per hr.	WEIGHT IN POUNDS: Normal	WEIGHT IN POUNDS: Total Loss	WEIGHT IN POUNDS: Rate of Loss lbs. per hr.
	Eff. T.— 88.5	McConnell	10:50	2:05	3.25	99.2	1.3	2.17	0.599	84	18	2.17	8.31	138.88	1.28	0.394
41	D. B. — 96.9	Flinn	10:50	2:05	3.25	98.8	1.0	2.17	0.462	80	12	2.17	5.54	97.15	1.00	0.308
	W. B. — 85.7	Fulton	10:50	2:00	3.17	98.2	1.0	2.17	0.462	76	6	2.17	2.77	141.09	1.91	0.602
7–18	R. H. — 63%								Average 0.51				Average 5.5			Average 0.44
	Eff. T.— 83.2	McConnell	10:40	2:00	3.33	98.4	0.9	2.83	0.318	72	6	2.83	2.12	138.63	0.80	0.243
42	D. B. — 97.2	Flinn	10:40	2:00	3.33	99.0	0.6	2.83	0.212	78	9	2.83	3.18	97.50	1.38	0.414
	W. B. — 77.0	Fulton	10:40	1:50	3.17	97.9	0.8	2.83	0.283	64	10	2.83	3.53	140.59	0.97	0.306
7–20	R. H. — 40.1%								Average 0.27				Average 2.9			Average 0.32
	Eff. T.— 81.0	McConnell	11:15	2:20	3.09	98.4	0.6	3.00	0.200	72	6	3.00	2.00	139.25	0.88	0.285
43	D. B. — 96.9	Flinn	11:15	2:25	3.17	99.0	0.6	3.00	0.200	74	10	3.00	3.33	96.88	0.72	0.227
	W. B. — 72.8															
7–21	R. H. — 31.0%								Average 0.20				Average 2.7			Average 0.26
	Eff. T.— 95.1	McConnell	10:45	1:50	3.09	98.8	2.4	3.00	0.800	78	42	3.00	14.00	140.63	3.00	0.970
44	D. B. — 97.1	Flinn	10:45	1:45	3.00	99.1	1.9	3.00	0.630	70	33	3.00	11.00	98.50	1.00	0.333
	W. B. — 94.6	Fulton	10:45	1:42	2.95	98.6	3.4	3.00	1.130	76	55	3.00	18.33	139.38	4.38	1.487
7–25	R. H. — 91.2%								Average 0.85				Average 14.4			Average 0.93
	Eff. T.— 89.3	McConnell	10:50	2:00	3.17	98.4	1.3	3.17	0.411	72	30	3.17	9.48	142.94	1.88	0.594
45	D. B. — 99.0	Flinn	10:50	2:00	3.17	99.0	0.7	3.17	0.221	71	9	3.17	2.84	100.81	0.94	0.313
	W. B. — 86.2	Fulton	10:45	2:00	3.25	98.0	1.0	3.17	0.317	72	24	3.17	7.58	140.00	1.88	0.578
8–10	R. H. — 60.0%								Average 0.32				Average 6.6			Average 0.50
	Eff. T.— 87.4	McConnell	11:00	2:10	3.17	98.0	1.5	3.00	0.500	73	11	3.00	3.67	141.81	1.13	0.356
46	D. B. — 88.3	Flinn	11:00	2:00	3.00	98.7	0.7	3.00	0.233	78	—3	3.00	—1.00	100.28	0.59	0.197
	W. B. — 87.0	Fulton	11:00	2:00	3.00	98.0	0.6	3.00	0.200	75	—6	3.00	—2.00	139.75	0.75	0.250
8–11	R. H. — 95.0%								Average 0.31				Average 0.2			Average 0.27

47	Eff. T.— 99.0	McConnell	10:50	12:30	1.67	98.5	2.8	1.42	1.97	78	66	1.42	46.6	141.57	3.44	2.060
	D. B. — 99.0	Flinn	10:50	12:30	1.67	99.2	3.2	1.72	1.86	80	43	1.72	25.0	100.50	2.12	1.270
	W. B. — 98.9	Fulton	10:50	12:35	1.75	98.0	4.2	1.75	2.40	68	51	1.75	29.1	139.88	3.63	2.075
8–15	R. H. — 99.9%								Average 2.08				Average 33.2			Average 1.80
48	Eff. T.— 82.7	McConnell	11:00	2:10	3.17	98.6	1.0	3.00	0.313	72	12	3.00	4.00	141.50	1.19	0.376
	D. B. — 99.0	Flinn	11:00	2:00	3.00	98.6	0.6	3.00	0.200	75	—2	3.00	—0.67	99.78	0.94	0.313
	W. B. — 75.1	Fulton	11:15	2:15	3.00	98.0	0.6	3.00	0.200	74	—4	3.00	—1.33	139.06	1.13	0.377
8–17	R. H. — 32.7%								Average 0.24				Average 0.7			Average 0.36
49	Eff. T.— 84.6	McConnell	11:00	2:10	3.17	98.6	1.0	3.00	0.333	72	12	3.00	4.00	142.06	1.31	0.413
	D. B. —101.9	Flinn	11:05	2:05	3.00	99.2	0.0	3.00	0.000	73	2	3.00	0.67	99.31	0.97	0.323
	W. B. — 77.3	Fulton	11:00	2:00	3.00	98.6	0.0	3.00	0.000	69	4	3.00	1.33	139.00	1.50	0.500
8–18	R. H. — 33.1%								Average 0.11				Average 2.0			Average 0.41
50	Eff. T.— 91.9	McConnell	11:10	2:00	2.83	98.5	2.9	2.83	1.023	80	62	2.83	21.88	143.38	4.13	1.460
	D. B. —102.0	Flinn	11:10	2:00	2.83	99.2	1.3	2.83	0.459	71	24	2.83	8.47	99.22	1.84	0.650
	W. B. — 89.0															
8–22	R. H. — 60.1%								Average 0.74				Average 15.2			Average 1.06
51	Eff. T.— 93.2	McConnell	11:00	2:00	3.00	98.6	2.2	3.00	0.733	72	54	3.00	18.00	143.75	3.72	1.240
	D. B. —104.1	Flinn	10:55	2:00	3.08	99.1	1.4	3.00	0.467	75	14	3.00	4.67	99.44	2.16	0.700
	W. B. — 90.4															
8–24	R. H. — 60%								Average 0.60				Average 11.3			Average 0.97
52	Eff. T.— 85.3	McConnell	11:00	2:00	3.00	98.6	0.8	3.00	0.267	72	16	3.00	5.33	143.25	1.37	0.460
	D. B. —101.9	Flinn	11:00	2:00	3.00	99.1	0.3	3.00	0.100	74	1	3.00	0.33	99.28	1.00	0.333
	W. B. — 78.5															
8–25	R. H. — 37.0%								Average 0.18				Average 2.8			Average 0.40
53	Eff. T.—101.3	McConnell	10:50	12:05	1.25	98.4	2.36	0.92	2.574	72	64	0.92	69.8	143.63	2.88	2.303
	D. B. —102.1	Flinn	10:50	12:05	1.25	99.0	2.28	0.92	2.487	71	39	0.92	42.7	99.89	1.32	1.055
	W. B. —101.2	Fulton	10:50	12.05	1.25	98.0				73				138.69	2.88	2.342
8–29	R. H. — 99.2%								Average 2.53				Average 56.3			Average 1.90
54	Eff. T.— 85.2	McConnell	11:00	2:00	3.00	98.2	1.1	3.00	0.367	72	17	3.00	5.67	142.78	1.28	0.427
	D. B. —104.0	Flinn	11:00	2:00	3.00	98.8	0.7	3.00	0.233	71	6	3.00	2.00	99.56	1.06	0.353
	W. B. — 77.3	Fulton	11:00	2:00	3.00	98.6	0.2	3.00	0.067	83	—5	3.00	—1.67	136.81	1.50	0.500
8–31	R. H. — 30%								Average 0.22				Average 2.00			Average 0.43

TABLE 1 (CONTINUED)

Test No. Date 1922	Test Conditions	Subjects	TIME: Entered Chamber	TIME: Left Chamber	TIME: In Chamber (hrs.)	RECTAL TEMPERATURE: Normal	Total Rise	Time	Rate of Rise deg. per hr.	PULSE RATE IN BEATS PER MIN.: Normal	Total Increase	Time	Rate of Increase per hr.	WEIGHT IN POUNDS: Normal	Total Loss	Rate of Loss lbs. per hr.
	Eff. T.—104.0	McConnell	10:55	11:35	0.67	98.3	3.7	0.67	5.550	72	78	0.67	117.0	143.63	3.50	5.250
55	D. B. —104.0	Flinn	10:55	11:50	0.92	99.1	3.4	0.67	5.022	71	66	0.67	99.0	99.00	1.88	2.042
	W. B. —104.0	Fulton	10:55	11:45	0.83	97.8				74	55	0.67	82.5	138.31	3.00	3.617
9-1	R. H. —100%								Average 5.29				Average 99.5			Average 3.64
	Eff. T.—106.0	McConnell	11:00	11:45	0.75	98.5	2.7	0.75	3.600	78	87	0.75	116.0	143.38	2.25	3.000
56	D. B. —106.0	Flinn	11:00	11:45	0.75	99.0	3.1	0.75	4.133	73	74	0.75	98.7	98.47	....	....
	W. B. —106.0	Fulton	11:00	11:45	0.75	98.2	3.8	0.75	5.067	70	75	0.75	100.0	137.25	1.92	2.560
9-12	R. H. —100%								Average 4.27				Average 104.9			Average 2.78
	Eff. T.— 86.5	McConnell	11:00	2:00	3.00	98.6	0.5	3.00	0.17	78	—4	3.00	—1.3	144.31	1.37	0.460
57	D. B. —106.1	Flinn	11:00	2:00	3.00	99.3	0.1	3.00	0.03	82	—2	3.00	—0.7	98.19	1.06	0.353
	W. B. — 78.8	Fulton	11:00	2:00	3.00	98.6	0.1	3.00	0.03	70	10	3.00	3.3	138.56	1.37	0.460
9-14	R. H. — 30%								Average 0.08				Average 0.4			Average 0.43
	Eff. T.— 94.9	McConnell	10:50	1:50	3.00	98.4	2.6	3.00	0.866	78	42	3.00	14.0	145.14	2.63	0.870
58	D. B. —106.0	Flinn	10:50	1:50	3.00	99.3	1.7	3.00	0.566	79	20	3.00	6.6	98.19	3.00	1.000
	W. B. — 92.2															
9-15	R. H. — 60%								Average 0.72				Average 10.3			Average 0.93
	Eff. T.— 87.5	McConnell	10:40	1:50	3.17	98.6	1.1	2.83	0.388	72	12	2.83	4.24	143.90	1.28	0.404
59	D. B. —108.0	Flinn	10:40	1:40	3.00	99.0	0.8	2.83	0.282	76	6	2.83	2.12	99.38	1.06	0.353
	W. B. — 79.8	Fulton	10:50	1:55	3.09	98.3	0.7	2.83	0.247	85	—5	2.83	—1.76	138.88	1.50	0.485
9-21	R. H. — 29.9%								Average 0.31				Average 1.5			Average 0.41
	Eff. T.—107.8	McConnell	10:50	11:35	0.75	98.6	3.9	0.75	5.200	90	78	0.67	117.0	142.75	2.12	2.825
60	D. B. —107.8	Flinn	10:50	11:40	0.83	98.8	3.5	0.77	4.561	76	..	...	....	100.25	1.88	2.266
	W. B. —107.7	Fulton	10:50	11:40	0.83	98.4	5.4	0.77	7.040	84	75	0.67	112.4	138.19	2.63	3.170
9-22	R. H. —100%								Average 5.60				Average 114.7			Average 2.754

61	Eff. T.— 96.9	McConnell	11:00	2:00	3.00	98.6	2.2	2.00	1.100	78	42	2.00	21.00	142.75	2.31	0.770
	D. B. —108.4	Flinn	11:00	2:00	3.00	98.6	2.5	2.00	1.250	75	26	2.00	13.00	99.75	1.69	0.563
	W. B. — 94.5	Fulton	11:00	2:00	3.00	98.2	2.9	2.00	1.450	70	52	2.00	26.00	138.38	2.88	0.960
9–26	R. H. — 60%								Average 1.27				Average 20.0			Average 0.77
62	Eff. T.— 88.9	McConnell	11:00	2:00	3.00	98.4	1.4	2.00	0.700	84	0	2.00	0.00	142.31	1.44	0.480
	D. B. —110.0	Flinn	11:00	2:00	3.00	98.8	0.6	2.00	0.300	71	6	2.00	3.00	100.75	1.06	0.353
	W. B. — 81.6	Fulton	11:00	2:00	3.00	98.0	0.7	2.00	0.350	74	6	2.00	3.00	137.81	1.38	0.460
9–28	R. H. — 30%								Average 0.45				Average 2.0			Average 0.43
63	Eff. T.—110.0	McConnell	11:00	11:45	0.75	98.6	3.4	0.67	5.100	84	84	0.75	112.00	141.88	2.69	3.585
	D. B. —110.0	Flinn	11:00	11:45	0.75	99.0	2.5	0.67	3.748	84	76	0.75	101.30	100.63	2.12	2.840
	W. B. —110.0	Fulton	11:00	11:45	0.75	98.0	3.0	0.67	4.500	70	79	0.75	105.10	137.31	2.31	3.080
9–29	R. H. —100%								Average 4.45				Average 106.1			Average 3.17
64	Eff. T.— 98.0	McConnell	10:30	12:10	1.67	98.6	2.4	1.50	1.600	72	48	1.50	32.00	139.25	1.12	0.670
	D. B. —110.1	Flinn	10:30	1:00	2.50	98.7	2.3	1.50	1.533	72	25	1.50	16.67	101.38	1.56	0.624
	W. B. — 95.7	Fulton	10:30	1:00	2.50	97.8	3.5	1.50	2.333	70	62	1.50	41.30	138.00	2.31	0.925
10– 3	R. H. — 60%								Average 1.82				Average 30.0			Average 0.74
65	Eff. T.—112.0	McConnell	10:47	11:16	0.48	98.6	2.2	0.366	6.015	72	78	0.366	213.0	140.31	3.06	6.380
	D. B. —112.0	Flinn	10:47	11:15	0.47	98.8	2.1	0.366	5.730	74	66	0.366	180.3	99.40	1.34	2.850
	W. B. —112.0	Fulton	10:47	11:16	0.48	98.3	3.8	0.467	8.140	80	80	0.366	218.5	137.63	2.19	4.560
10–10	R. H. —100%								Average 6.63				Average 203.9			Average 4.60
66	Eff. T.— 99.0	McConnell	10:50	12:45	1.92	98.4	2.6	1.42	1.83	75	51	1.42	36.0	141.38	1.60	0.833
	D. B. —112.0	Flinn	10:50	12:45	1.92	98.8	2.6	1.42	1.83	70	48	1.42	33.9	100.00	....	....
	W. B. — 96.7	Fulton	10:50	12:45	1.92	98.3	3.3	1.42	2.33	70	64	1.42	45.2	138.00	1.37	0.713
10–12	R. H. — 60%								Average 2.00				Average 38.4			Average 0.77
67	Eff. T.— 90.3	McConnell	10:45	1:45	3.00	98.6	1.1	3.00	0.367	84	6	3.00	2.00	140.69	1.56	0.520
	D. B. —112.0	Smith	10:45	1:45	3.00	98.7	1.2	3.00	0.400	76	12	3.00	4.00	120.00	1.38	0.460
	W. B. — 83.2	Fulton	10:45	1:45	3.00	98.6	0.7	3.00	0.233	80	8	3.00	2.67	138.00	2.00	0.667
10–17	R. H. — 30%								Average 0.33				Average 2.89			Average 0.55
68	Eff. T.— 91.4	McConnell	10:55	2:00	3.08	98.5	1.0	3.00	0.333	78	12	3.00	4.00	140.19	1.75	0.567
	D. B. —114.1	Smith	10:55	2:00	3.08	98.6	1.0	3.00	0.333	72	20	3.00	6.67	122.00	2.37	0.769
	W. B. — 84.6	Fulton	11:00	2:00	3.00	98.3	1.0	3.00	0.333	76	10	3.00	3.33	138.31	2.31	0.770
10–19	R. H. — 30%								Average 0.33				Average 4.7			Average 0.70

TABLE 1 (CONCLUDED)

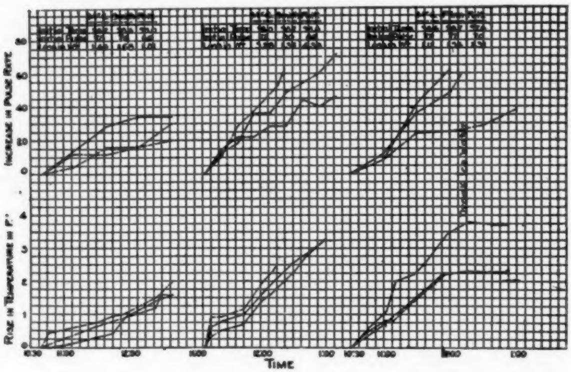
Test No. Date 1922	Test Conditions	Subjects	TIME: Entered Chamber	TIME: Left Chamber	TIME: In Chamber (hrs.)	RECTAL TEMPERATURE: Normal	RECTAL TEMPERATURE: Total Rise	RECTAL TEMPERATURE: Time	RECTAL TEMPERATURE: Rate of Rise deg. per h	PULSE RATE IN BEATS PER MIN.: Normal	PULSE RATE IN BEATS PER MIN.: Total Increase	PULSE RATE IN BEATS PER MIN.: Time	PULSE RATE IN BEATS PER MIN.: Rate of Increase per hr.	WEIGHT IN POUNDS: Normal	WEIGHT IN POUNDS: Total Loss	WEIGHT IN POUNDS: Rate of Loss lbs. per hr.
70	Eff. T.—101.3	McConnell	10:45	12:00	1.25	98.4	3.0	1.18	2.54	72	90	1.25	72.0	139.50	2.50	2.000
	D. B. —114.0	Smith	10:45	11:50	1.08	98.4	2.7	1.00	2.70	84	76	1.08	70.3	119.25	1.75	1.750
	W. B. — 99.3	Fulton	10:45	12:00	1.25	98.0	3.5	1.18	2.98	70	90	1.25	72.0	139.63	3.63	2.908
10-24	R. H. — 60%								Average 2.74				Average 71.1			Average 2.22
71	Eff. T.— 92.8	McConnell	10:50	1:50	3.00	98.2	1.1	2.50	0.440	72	18	2.50	7.20	140.38	1.88	0.626
	D. B. —116.0	Smith	10:50	1:40	2.83	98.2	1.7	2.50	0.680	72	24	2.50	9.60	119.13	1.63	0.574
	W. B. — 86.2	Fulton	10:50	1:50	3.00	98.2	1.1	2.50	0.440	83	13	2.50	5.20	139.63	2.19	0.730
10-26	R. H. — 30%								Average 0.52				Average 7.3			Average 0.64
72	Eff. T.—106.2	McConnell	10:50	11:35	0.75	98.6	3.0	0.72	4.18	72	96	0.750	128.0	140.75	2.12	2.826
	D. B. —120.0	Smith	10:50	11:20	0.50	98.4	2.2	0.45	4.90	80	60	0.367	163.5	119.63	0.89	1.778
	W. B. —104.6	Fulton	10:50	11:40	0.83	97.8	3.6	0.72	5.02	74	69	0.750	92.0	139.13	2.60	3.130
10-27	R. H. — 60%								Average 4.70				Average 127.8			Average 2.58
73	Eff. T.—102.9	McConnell	10:45	11:45	1.00	98.6	3.2	1.00	3.200	78	54	0.75	67.5	142.13	3.06	3.060
	D. B. —116.2	Smith	10:45	11:25	0.67	98.6	1.3	0.67	1.950	78	83	0.75	103.7	118.88	1.75	2.610
	W. B. —101.0	Fulton	10:45	11:30	0.75	98.0	1.9	0.67	2.850	78	52	0.75	65.0	139.31	2.88	3.839
10-31	R. H. — 60%								Average 2.67				Average 83.8			Average 3.17
74	Eff. T.— 95.2	McConnell	10:40	12:40	2.00	98.2	1.6	2.00	0.800	78	30	2.00	15.00	140.69	1.69	0.845
	D. B. —120.0	Smith	10:40	12:40	2.00	98.6	2.0	2.00	1.000	76	20	2.00	10.00	119.75	1.68	0.840
	W. B. — 89.2	Fulton	10:40	12:40	2.00	98.0	1.6	2.00	0.800	66	34	2.00	17.00	139.38	1.81	0.905
11-2	R. H. — 30%								Average 0.87				Average 14.0			Average 0.86
76	Eff. T.—101.8	McConnell	10:50	12:20	1.50	98.0	2.9	1.42	2.076	78	66	1.500	44.00	138.44	2.26	1.504
	D. B. —129.3	Smith	10:50	11:50	1.00	98.4	4.0	1.42	2.804	80	60	0.917	65.50	117.00	1.50	1.500
	W. B. — 97.5	Fulton	10:50	12:23	1.55	98.2	3.5	1.42	2.470	76	68	1.500	45.30	138.75	2.64	1.702
11-28	R. H. — 30.2%								Average 2.45				Average 51.6			Average 1.57

77	Eff. T.—113.9	McConnell	10:55	11:25	0.50	98.6	3.00	0.47	6.430	78	66	0.17	395.0	137.81	1.38	2.760
	D. B. —129.9	Smith	10:55	11:10	0.25	99.0	3.00	0.42	7.200	88	52	0.17	311.7	115.00	0.63	2.520
	W. B. —113.0	Fulton	10:55	11:20	0.42	98.0	3.00	0.42	7.200	70	83	0.17	497.0	140.19	2.10	5.000
12- 1	R. H. — 58.9%								Average 6.94				Average 401.2			Average 3.43
78	Eff. T.—101.1	McConnell	10:55	12:25	1.50	98.8	3.2	1.52	2.109	78	78	1.50	52.0	137.50	3.90	2.600
	D. B. —130.0	Smith	10:55	11:55	1.00	98.6	2.7	0.90	3.000	92	68	1.00	68.0	117.75	1.38	1.380
	W. B. — 96.4	Fulton	10:55	12:30	1.58	98.6	3.4	1.53	2.218	80	68	1.58	42.9	139.38	3.44	2.179
12- 5	R. H. — 30%								Average 2.44				Average 54.3			Average 2.05
79	Eff. T.—101.2	McConnell	11:10	1:10	2.00	98.0	3.3	1.92	1.709	72	46	1.27	36.2	138.38	3.28	1.640
	D. B. —146.0	Smith	11:10	12:20	1.16	98.2	2.4	1.07	2.250	80	62	1.27	48.8	119.75	1.35	1.167
	W. B. — 93.6	Fulton	11:10	1:10	2.00	98.6	3.3	1.87	1.741	86	..	...	...	140.13	4.38	2.190
12- 7	R. H. — 15%								Average 1.90				Average 42.5			Average 1.67
80	Eff. T.—106.4	McConnell	10:50	11:35	0.75	98.2	3.4	0.75	4.466	76	80	0.75	106.5	139.06	2.56	3.410
	D. B. —157.0	Smith	10:50	11:15	0.42	98.6	2.2	0.42	5.230	80	62	0.33	186.1	119.63	1.25	2.980
	W. B. —100.4	Fulton	10:50	11:35	0.75	97.9	1.8	0.42	4.167	80	60	0.75	80.0	140.87	5.33	7.110
12- 8	R. H. — 15%								Average 4.60				Average 124.2			Average 4.50
81	Eff. T.— 95.1	McConnell	11:05	1:35	2.50	99.0	1.3	2.50	0.500	84	24	2.50	9.6	137.75	2.50	1.000
	D. B. —144.7	Smith	11:05	1:35	2.50	98.6	2.5	2.50	1.000	80	40	2.50	16.0	118.00	2.63	1.052
	W. B. — 81.8															
12-14	R. H. — 5%								Average 0.75				Average 12.8			Average 1.03
82	Eff. T.— 90.3	McConnell	10:45	1:45	3.00	98.4	0.6	3.00	0.200	78	6	3.00	2.0	138.19	0.312	0.104
	D. B. — 90.3	Smith	10:45	1:45	3.00	98.8	1.2	3.00	0.383	80	4	3.00	1.3	119.75	1.375	0.458
	W. B. — 90.3															
12-15	R. H. —100%								Average 0.29				Average 1.7			Average 0.28
83	Eff. T.— 85.0	McConnell	10:40	1:40	3.00	98.7	0.06	3.00	0.020	78	6	3.00	2.0	136.88	0.375	0.125
	D. B. — 85.0	Smith	10:40	1:40	3.00	98.9	0.30	3.00	0.100	72	0	3.00	0	119.38	0.063	0.021
	W. B. — 85.0															
12-21	R. H. —100%								Average 0.06				Average 1.0			Average 0.073
84	Eff. T.— 80.0	McConnell	11:00	2:00	3.00	99.2	—0.6	3.00	—0.2000	84	0	3.00	0	....	....	....
	D. B. — 80.0	Smith	11:00	2:00	3.00	98.6	+0.25	3.00	+0.0830	76	—11	3.00	—3.667	....	....	....
	W. B. — 80.0															
12-22	R. H. —100%								Average —0.058				Average —1.833			

Test 74—95.2° E.T. 120° D.B., 89.2° W.B., 30% R.H.

Test 79—101.2° E.T. 146.0° D.B., 93.6° W.B., 15% R.H.

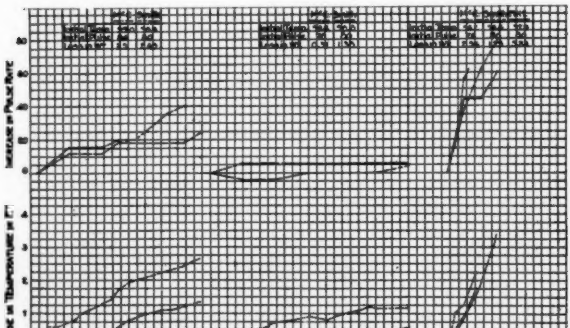
Test 64—98.0° E.T. 110.1° D.B., 95.7° W.B., 60% R.H.



Test 81—95.1° E.T. 144.7° D.B., 81.8° W. B., 5% R.H.

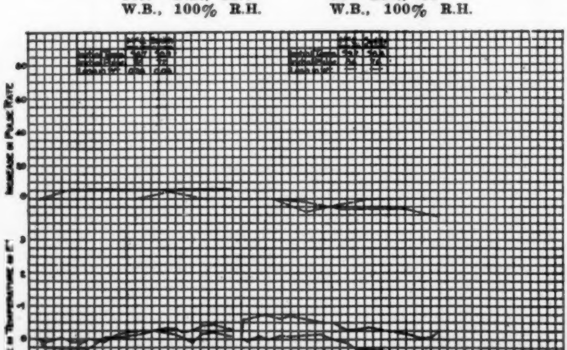
Test 82—90.3° E.T. 90.3° D.B., 90.3° W.B., 100% R.H.

Test 80—106.4° E.T. 157.0° D.B., 100.4° W.B., 15% R.H.



Test 83—85.0° E.T. 85.0° D.B., 85.0° W.B., 100% R.H.

Test 84—80° E.T. 80° D.B., 80° W.B., 100% R.H.



No. 655

DETERMINING LINES OF EQUAL COMFORT

By F. C. Houghten[1]. Pittsburgh, Pa. (Member)
and
C. P. Yagloglou[2], Pittsburgh, P. (Non-Member)

THE bodily feeling of warmth is not due alone to the temperature indicated by the dry bulb thermometer, neither does it depend solely upon the wet bulb temperature. Dry air at a relatively high temperature may feel cooler than air of considerably lower temperature with high moisture content.

Human comfort or discomfort depend largely on the relation between the rate of heat production and dissipation. By the process of metabolism heat is constantly generated within the body, while on the other hand, loss of heat is constantly occurring from the surface of the body by radiation, convection and evaporation. To maintain a constant body temperature the loss of heat must equal the heat produced. It is therefore apparent that any interference with the elimination of heat from the body is accompanied by a rise in temperature and a feeling of discomfort.

There are three principal factors affecting loss of body heat:

1. Temperature.
2. Humidity.
3. Air motion.

As the temperature of the air and surrounding objects rises, the loss of heat by convection and radiation decreases. When the temperature reaches that of the body, the loss by radiation and convection ceases. Finally as the air temperature exceeds that of the body, heat passes from the air to the body.

If, on the other hand, the relative humidity is increased the heat loss by evaporation decreases. If while the dry bulb temperature increases, the wet bulb temperature decreases sufficiently, the increase in the loss of heat by evaporation may be made equal to the decrease in the loss of heat by radiation and convection, resulting in no change in the thermal state of the body temperature or comfort.

[1-2] Research Engrs., A. S. H.-V. E. Research, Laboratory, U. S. Bureau of Mines.
Paper presented at the Annual Meeting of the American Society of Heating and Ventilating Engineers, Washington, D. C., January, 1923.

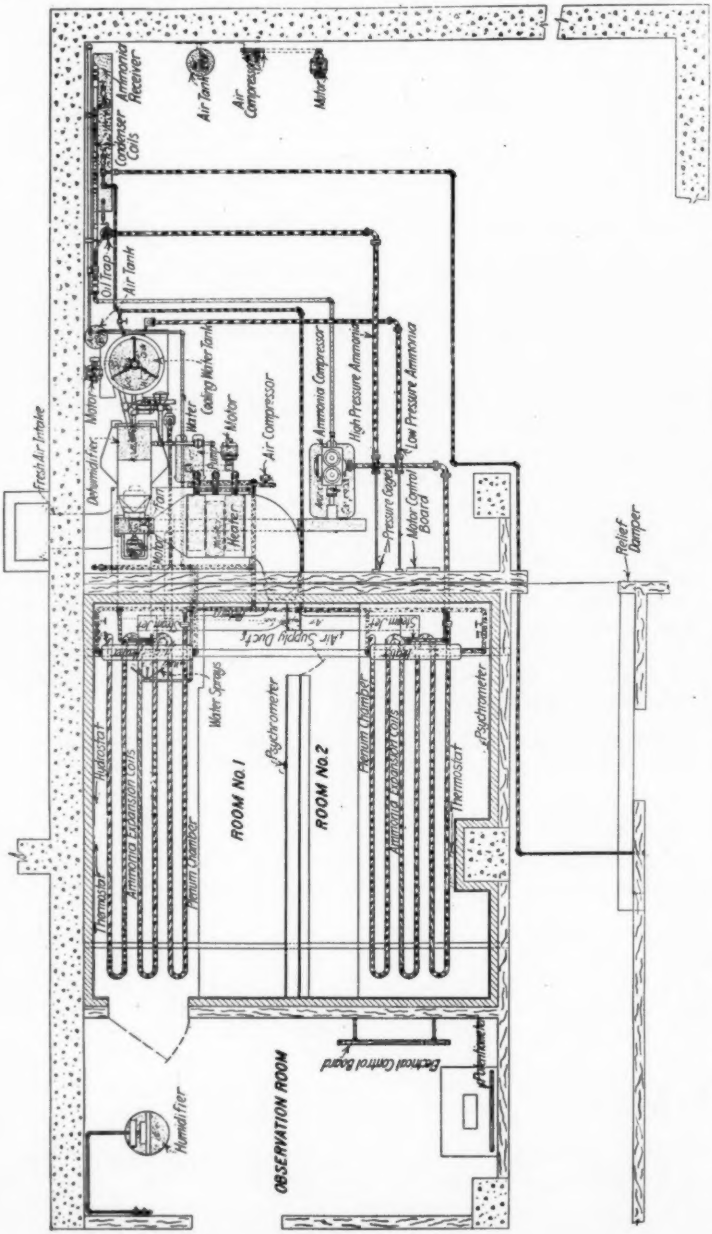


FIG. 1. PLAN OF PSYCHROMETRIC CHAMBER AND AUXILIARY EQUIPMENT IN ADJOINING ROOMS

From the above, it is concluded that there must necessarily be certain combinations of temperatures and humidities, which produce the same total body heat loss by radiation, convection and evaporation and therefore the same feeling of comfort or discomfort. Lines passing through such air conditions may be called equal comfort lines, The fact is further substantiated by the general experience of heating engineers in observing that the lower the humidity the higher the temperature required for the same degree of comfort. Determining these equal comfort lines for various temperatures, humidities and

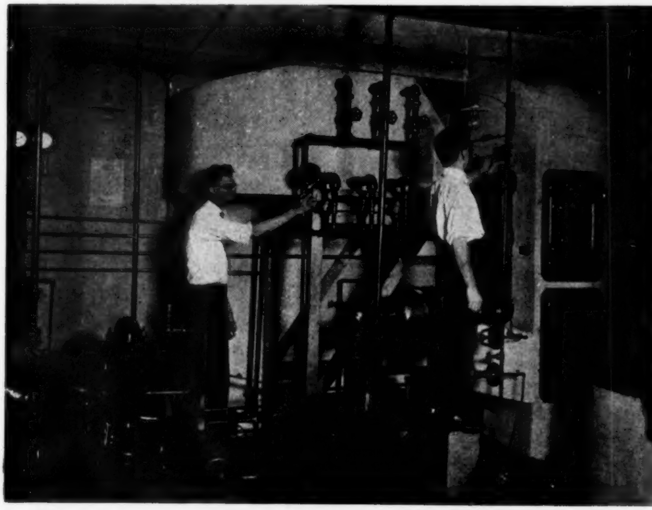


FIG. 2. AIR CONDITIONING APPARATUS

air velocities is the object of an important investigation being carried on at the Research Laboratory. The determination of these equal comfort lines for still air conditions is completed and the results are given in this paper.

The investigation of the effect of the various air velocities on the equal comfort lines is being continued at the Laboratory and a report on this work will be rendered in a subsequent paper.

EQUIPMENT

The Research Laboratory is ideally equipped for conducting this work. There are two psychrometric rooms 8 x 17 x 10 ft., insulated with 4 in. of cork board, besides the regular building construction, and plastered on the inside. The temperature, humidity and air motion in the chambers can be controlled independently of each other. The air conditioning apparatus is located in an adjoining room and

includes a humidifier or dehumidifier, heating and cooling coils, an ammonia compressor, cooling tank, blowers, etc. The plan of these rooms, with the location of the air conditioning equipment and other accessories, is shown in Fig. 1.

Re-circulated air, air from the corridors of the buildings or outside air, or any combination of the three is drawn through a tower of sprays, which acts as air washer and humidifier or dehumidifier, depending upon the temperature of the spray water. The air then passes through a fan, over heaters and divides into two ducts supply-

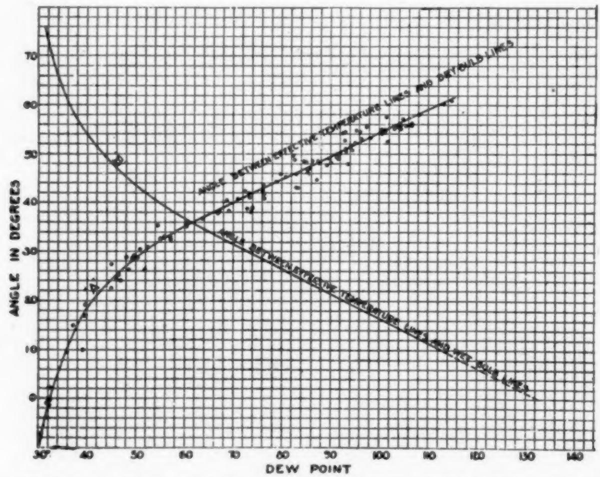
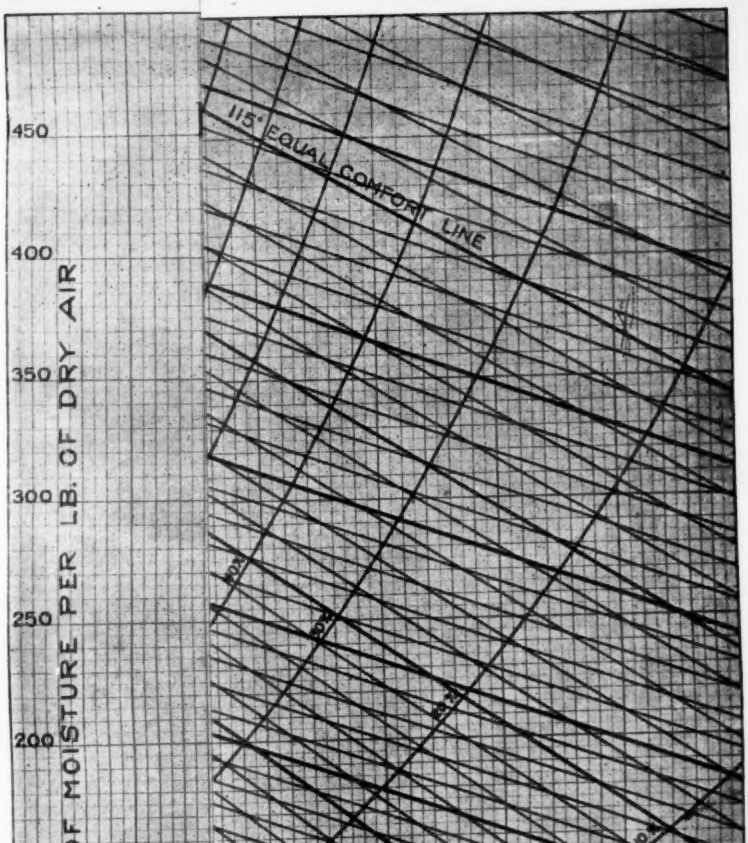


FIG. 3. CURVE *A*, ANGLE BETWEEN EQUAL COMFORT LINES AND DRY BULB LINES AS DETERMINED BY TEST. CURVE *B*, ANGLE BETWEEN EQUAL COMFORT LINES AND WET BULB LINES

ing the two rooms. Some of the air can be made to bypass the heaters into the duct, leading to the first room. The ducts discharge the air into plenum chambers, 5 x 17 x 3 ft., at the top of each room. In the stream of air leaving the duct there is a set of water-sprays and a steam jet. A 60 sq. ft. radiator is located in the plenum chamber of each room in such a way that the incoming air passes over it. Each plenum chamber also contains refrigerating coils. The air enters the rooms from the plenum chambers through a large number of small holes widely distributed. This arrangement gives negligibly small air motion in the rooms.

The temperature of the water used to wash and humidify or dehumidify the air supplied to the rooms, is thermostatically controlled by means of automatic air valves which admit either steam or water from a cooling tank containing refrigerating coils. A thermostat in each room controls either the steam supplied to the main heaters or that supplied to the small radiators in the plenum chambers. The



450
400
350
300
250
200
150
100
50
CRAINS OF MOISTURE PER LB. OF DRY AIR
115° EQUAL COMFORT LINE
30
35
30
130
140
150
160
F.C. Houghten
C.P. Yaglogloa

'ERIMPOSED

water-spray in the first room is controlled by a hygrostat. The amount of air admitted to either room is controlled by dampers. This arrangement gives very flexible and independent control of all the factors entering into the air conditions of either room.

The range of temperature and humidity attainable is very great but is limited by the capacity of the heaters and ammonia compressor. Saturated temperatures which a human being can stand for only a few minutes can be obtained without any difficulty. Low humidities with high temperatures are limited, however, by reason of the fact

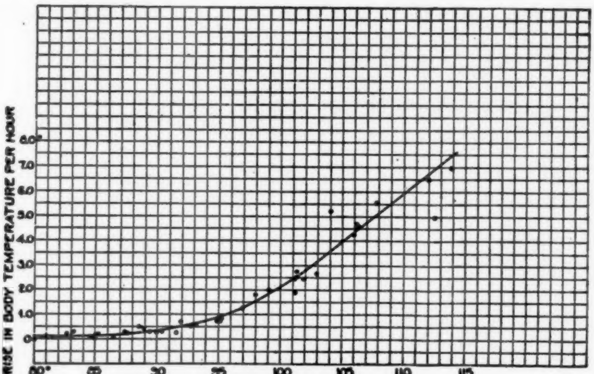


FIG. 5. CURVE SHOWING RELATION BETWEEN EFFECTIVE TEMPERATURE AND RATE OF RISE IN BODY TEMPERATURE

that the air before passing over the heaters must be cooled to the dew-point of the desired condition. With recirculated air it is not possible to obtain a very low humidity with high temperature. However, by taking outside air on a cold day, a very low dew-point can be obtained, making possible very low humidities at high temperatures.

METHOD OF CONDUCTING TESTS

In conducting tests to determine the equal comfort lines, the atmospheric condition in the second psychrometric room is accurately maintained at a relatively high dry bulb temperature and low relative humidity. The atmospheric condition of the first psychrometric room is brought to a lower dry bulb temperature and higher relative humidity, such that judges passing from one room to the other decide that it feels slightly cooler than the second room.

The dry or wet bulb temperature, or both, of the first chamber are then allowed to slowly rise while the judges pass back and forth from one chamber to the other, each time recording their judgment as to the relative feeling of warmth of the two conditions. As the temperature or humidity of the first chamber increases, a point is reached

where the observers judge that the sense of warmth in the two chambers is the same. As the temperature or humidity of the first chamber continues to rise a point is reached where the first chamber is the warmer of the two, although its dry bulb temperature may be as much as 50 deg. lower than that of the second room. The change in wet bulb or dry bulb temperature of the first room, from the point where the judges agree that it is the cooler of the two, to the point where they agree that it is the warmer of the two, seldom exceeds 1 deg. In fact, it is usually considerably less than 1 deg. Table 1 shows the results of an actual test, in which three judges recorded their independent feeling. It will be seen that from the point where the three judges agreed that the first room was cooler, to the point where they agreed that it was the warmer of the two, there is a difference of only 0.8 deg. dry bulb and 0.9 deg. wet bulb.

TABLE 1. SAMPLE DATA OF TESTS TO DETERMINE EQUIVALENT CONDITIONS

First Room		Second Room		Observer's Feelings		
D.B	W.B.	D.B.	W.B.	*A*	*B*	*C*
95.8	87.4	115.0	82.8	Cooler	Dec. Cooler	Cooler
96.2	87.6	115.0	82.8	Cooler	Cooler	Cooler
96.6	87.9	115.0	82.7	Sl. Cooler	Cooler	Sl. Cooler
96.8	88.1	115.0	82.8	Sl. Cooler	Sl. Cooler	Sl. Cooler
96.9	88.2	115.1	82.8	Same	Sl. Cooler	Sl. Cooler
97.0	88.6	115.0	82.8	Same	Same	Same
97.3	88.8	115.0	82.8	Sl. Warmer	Sl. Warmer	Same
97.6	89.0	115.0	82.8	Sl. Warmer	Sl. Warmer	Sl. Warmer
98.0	89.3	115.1	82.7	Sl. Warmer	Sl. Warmer	Warmer
98.8	89.5	115.2	82.8	Dec. Warmer	Dec. Warmer	Dec Warmer

Actual observations taken in one of 440 tests made to determine equal comfort lines. The two equal comfort conditions are represented by points *D* and *E* on the Psychrometric Chart, Fig. 10.

DATA AND RESULTS

Four hundred and forty similar tests have been made and the corresponding lines drawn on the standard psychrometric chart. The angles between these lines and the dry bulb temperature lines are plotted against the dew-point at their intersection with the saturation curve in Fig. 3. It will be seen that these points form a very definite graph approximating a parabolic curve. Of the 440 lines experimentally determined there are very few whose angle with the dry bulb lines fall very far from the curve drawn through their average.

A standard psychrometric chart with the equal comfort lines superimposed is shown in Fig. 4. These lines were determined by taking the angle which they form with the dry bulb lines from Curve *A*, Fig. 3.

It will be seen that they form a series of lines which are not parallel among themselves but approach being parallel to the wet bulb lines at high temperatures, and dry bulb lines at low temperatures. This is in accordance with expectations. At high temperatures one perspires freely and the skin approaches the condition of a wet bulb,

while at low temperatures the surface of the body is relatively dry and therefore it is little cooled by evaporation. At low temperatures moisture in the air allows the clothing to become moist and therefore to be a better conductor of heat.

Curve *B* in Fig. 3 shows the angles between wet bulb lines and equal comfort lines plotted against dew-points. It will be observed that at low dew-point temperatures dry bulb temperature is of greater importance, while at high dew-point temperature wet bulb temperature is the chief factor in determining comfort.

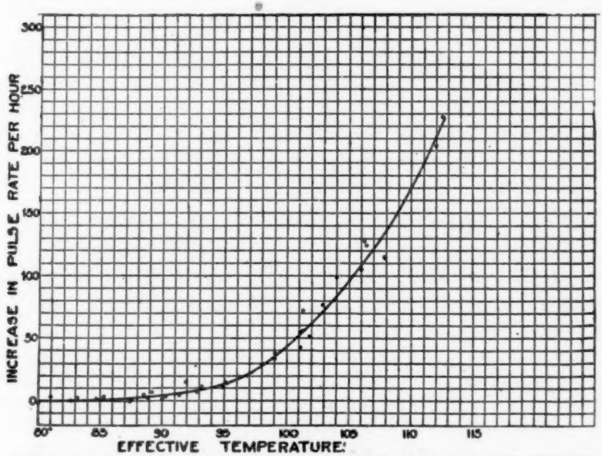


FIG. 6. CURVE SHOWING RELATION BETWEEN EFFECTIVE TEMPERATURE AND RATE OF PULSE INCREASE

Referring again to Curve *B*, Fig. 3, and extrapolating beyond the limits of the experimental data, it is found that the angle between wet bulb and equal comfort lines becomes zero at a temperature of dew-point 132 deg. Therefore at this temperature the wet bulb lines are also equal comfort lines, or comfort depends upon wet bulb temperature alone. This could not be verified by test for the reason that it is impossible for the investigators to endure the conditions long enough to determine lines above 115 deg. saturated.

Below 62 deg. the angle between the equal comfort lines and the dry bulb lines diminishes rapidly, approaching zero at 32 deg. For all points on the 32 deg. equal comfort line, comfort is independent of wet bulb temperature or relative humidity.

Below 32 deg. a marked reversal in the direction of the equal comfort lines occurs. Therefore for such conditions the higher the wet bulb temperature or relative humidity the greater the cooling effect of the air. This fact, while not generally accepted, receives ample verification from the experience of people who have dwelt in cold climates.

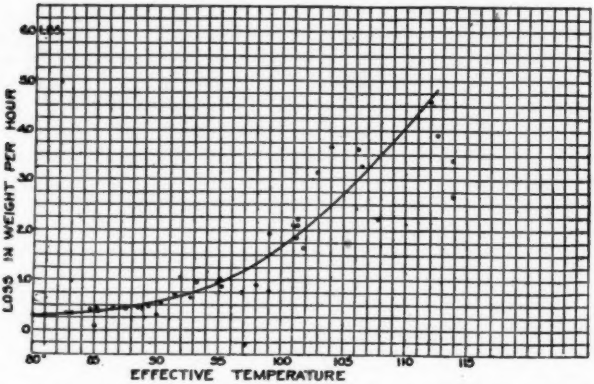


FIG. 7. CURVE SHOWING RELATION BETWEEN EFFECTIVE TEMPERATURE AND LOSS IN WEIGHT BY PERSPIRATION

The equal comfort lines as drawn in the chart are straight. While within the limits of experimental error this is true, there may possibly be a slight curvature, which will, however, in no case affect any point on any line, within the region of the chart investigated, more than 0.5 deg.

Passing from left to right on the chart one ascends the dry bulb temperature scale. The fact has long been recognized that dry bulb temperature does not alone determine a person's feeling of warmth. Rising diagonally from left to right the wet bulb temperature scale is followed. The so-called wet bulb temperature does not indicate the temperature of the air but rather that of a wet body when air moves around it sufficiently fast. This temperature scale is determined by the dry bulb temperature at the point where the two lines intersect the saturation curve.

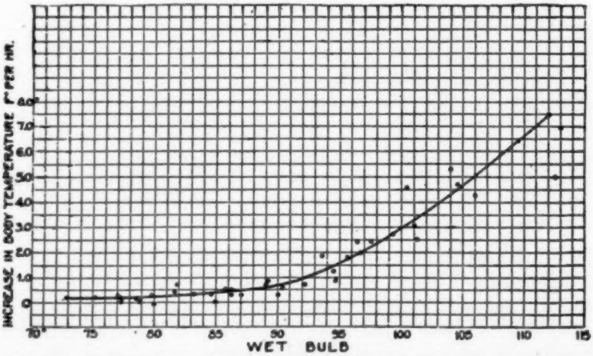


FIG. 8. CURVE SHOWING RELATION BETWEEN WET BULB TEMPERATURE AND RATE OF RISE IN BODY TEMPERATURE

It has already been shown that the wet bulb temperature does not alone indicate a person's feeling of warmth. While neither the dry nor wet bulb temperature indicate the sense of warmth the equal comfort lines do give such an index. For example, if one passes from any point on the equal comfort line intersecting the saturation curve at 90 deg. to any point on the equal comfort line intersecting the saturation curve at 91 deg., an equal increase in warmth will be noticed, regardless of the path followed. This series of equal comfort lines can therefore be thought of as a temperature scale which, unlike the wet bulb and dry bulb scale, determines accurately the feeling of warmth. The best way to fix the numerical value of the new scale is

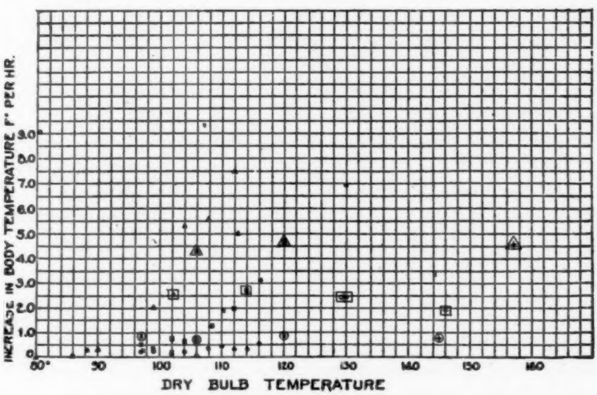


FIG. 9. CURVE SHOWING RELATION BETWEEN DRY BULB TEMPERATURE AND RATE OF RISE IN BODY TEMPERATURE

to adopt the value common to all scales at their intersection on the saturation curve.

As has already been said and will be further verified, this temperature scale accurately indicates one's feeling of warmth and determines the physiological reactions of the body. In other words, it entirely determines the effects produced upon the body. It may therefore be called a scale of effective temperature. The writers are cognizant of the fact that this so-called effective temperature scale neither indicates the temperature of the air nor that of any other object associated with the air, and that the use of this term may not be in entire accordance with accepted scientific terminology. However, they are convinced that such a temperature scale will be of sufficient value in the art of heating and ventilation to justify this departure.

Incidentally in parallel with this work the Research Laboratory in co-operation with the U. S. Bureau of Mines and the U. S. Public Health Service is investigating the physiological reactions of men to various temperature conditions. A report of this work has already been presented at this meeting. (JOURNAL A.S.H.-V.E. March, 1923.)

After determining the effective temperature scale by sense reactions alone it is gratifying to find that the physiological reactions as determined in the investigation previously referred to, also follow the new temperature scale as closely as could be expected, considering the difficulties involved in determining physiological reactions. In Fig. 5, the rise in body temperature is plotted against effective temperature. The data was taken from Table 1 of the report previously rendered. The points follow the curve very closely with the exception of two tests. Equally good results are obtained by plotting increase in pulse rate hour against effective temperature, in Fig. 6.

In Fig. 7, the loss in weight per hour by perspiration is plotted against effective temperature. The points do not plot as well as in the former two curves for the reason that the loss in weight was not determined as accurately as the increase in temperature and pulse rate.

The rise in body temperature plotted against wet bulb temperature is shown in Fig. 8. Comparing this curve with that in Fig. 5 it is noted that the rise in temperature does not plot as well against wet bulb as it does against effective temperature. A similar comparison can be shown in favor of effective temperature by plotting increase in pulse rate and loss in weight per hour against wet bulb temperature.

By plotting the rise in body temperature against dry bulb temperature in Fig. 9, the folly in the use of dry bulb temperature in determining comfort can be seen. In this curve, circles represent tests at 30 per cent relative humidity, squares represent tests at 60 per cent relative humidity, triangles, tests at 100 per cent relative humidity, and crosses, tests at humidities below 25 per cent. Distinct curves can be drawn through each of these three series of similar points.

Large triangles are drawn around points of test at 106 deg. effective temperature. Squares are drawn around points of tests made at 101 deg. effective temperature and circles around tests made at 95 deg. effective temperature. All tests in each of these three series produce about the same rise in body temperature regardless of dry bulb temperature.

To further emphasize the importance of effective temperature over the wet bulb temperature in determining comfort, some experimental facts are graphically shown on the psychrometric chart, Fig. 10. Conditions for the test are shown by points on the chart. The number of the test, increase in pulse rate, rise in body temperature and loss in weight per hour are given in the rectangles around the points. Three series of tests are given which are very close to the three effective temperature lines drawn. Tests Nos. 56, 72 and 80, falling near the 106 deg. effective temperature line, produce increases in pulse rate of 105, 128 and 124 respectively. Temperature rises and loss in weights per hour show a similar uniformity.

While these physiological reactions vary considerably among themselves, they are of a different order of magnitude from those found

in the next lower series, that is, near the 101 deg. effective temperature line. Near this line are four tests, Nos. 53, 76, 78 and 79, which give similar physiological reactions. Again on the 95 deg. effective temperature line we have four tests which give similar physiological reactions but quite different from those of the other two series.

Test No. 53 is 1 deg. wet bulb higher than test No. 80, but 5 deg. effective temperature lower. It is found that test No. 80 gives an increase in pulse rate of 124 and temperature rise of 4.6 deg., while test No. 53 gives an increase in pulse rate of 56.3 and temperature rise of 2.53 deg., which shows clearly that the physiological reactions in this region of the chart depend on effective temperature and not

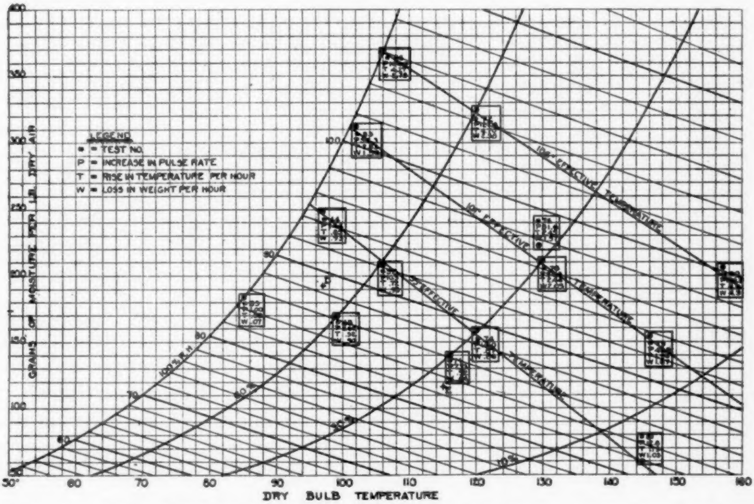


FIG. 10. PSYCHROMETRIC CHART, SHOWING RELATION BETWEEN PHYSIOLOGICAL REACTIONS AND EFFECTIVE TEMPERATURE, WET BULB TEMPERATURE AND DRY BULB TEMPERATURE, RESPECTIVELY

on wet bulb temperature. When the effective temperature is lower the physiological reactions are less marked despite the fact that the wet bulb temperature is higher.

Tests Nos. 44 and 79 similarly show the fact that the physiological reactions depend on effective temperature and not on wet bulb temperature. This is again brought out by tests Nos. 81, 74, 45, and 83. The lower the effective temperature, the less marked the physiological reactions regardless of wet bulb temperature.

APPLICATIONS

The applications of the psychrometric chart with equal comfort lines to the problems of heating and ventilation are many. The heat-

ing engineer is interested in the effective temperature lines within the comfort zone, that is, those temperatures and humidities which are met with in heating practice. To say that a room is heated to 70 deg. does not necessarily indicate that the room is heated to a comfortable temperature. With a very low humidity, 70 deg. fahr. may be too cold for comfort while 70 deg. fahr. with a very high humidity may be too warm.

It is not the purpose of this investigation to determine what effective temperature is best suited for true comfort. However, if results can be obtained that agree upon any point that gives true comfort the chart will indicate all other conditions which are equally comfortable. If 70 deg. dry bulb and 56 deg. wet bulb are accepted as being on the true comfort line then 63.5 deg. effective temperature will be the desired line. The maximum variation on this comfort line is from 63.5 deg. to 77 deg. dry bulb and from 47 deg. to 63.5 deg. wet bulb. However, the maximum variation is neither practical nor desirable.

In industrial plants having high temperatures the efficiency engineer inquires as to what combinations of dry and wet bulb temperatures can be endured with efficient work. The U. S. Public Health Service has determined from experience and statistics conditions which men can efficiently endure while doing various kinds of work. By accepting these points the chart will give any other conditions resulting in equally efficient work.

The air conditioning engineer is interested to know just what effect various changes in wet and dry bulb temperature will have on effective temperature. He can follow three distinct routes in producing a more desirable condition or any combination of the three.

1.—He can keep the dry bulb constant varying the wet bulb and dew-point.

2.—He can keep the wet bulb constant and vary the dry bulb and dew-point.

3.—He can keep the dew-point constant varying the dry and wet bulb.

4.—Any combination of the above three.

In any case the improvement will depend solely upon the change in effective temperature.

CONCLUSIONS

Comfort as determined by both sense and physiological reactions depends solely upon effective temperature. At 32 deg. the effective temperature line coincides with the dry bulb temperature line, hence in this particular case dry bulb temperature is the only factor in determining comfort.

In the comfort zone comfort depends equally on wet and dry bulb temperatures.

At about 132 deg. the effective temperature line coincides with the wet bulb temperature line when wet bulb temperature is the only factor.

Below 32 the effect of humidity is reversed. The lower the humidity the warmer the condition.

ACKNOWLEDGMENT

The writers wish to express their indebtedness to O. W. Armspach and W. H. Carrier for their many valuable suggestions and kindly criticism. Acknowledgment is also due to R. L. Lincoln and E. C. Emanuel for serving as judges and for their willingness to endure the undesirable heat conditions associated with the tests.

DISCUSSION

W. H. CARRIER: The development of the comfort chart is work which I feel the Society has greatly needed. We have had a committee on ventilation but we felt that the data available was inadequate in respect to the degree of comfort experienced in various temperatures and humidities. Dr. Hill has done a great deal of valuable work on this, but he probably did not have the facilities that we have at the Laboratory, and we now have data that is apparently perfectly reliable for still air. The effect of other conditions will be determined. I want to emphasize the importance of this research to us engineers. We want to know—this is discussed to some extent in the paper, but I want to emphasize it—we want to know, when we cool air, how much we are going to increase the comfort or whether we are going to increase it at all. For instance, suppose that we plan to cool an office or factory by simply saturating the air. We know it does not change the wet bulb. The argument might be put forward, since it does not change the wet bulb, it cannot increase the comfort, because the total heat in the air is the same and it has simply gone from sensible heat into latent heat.

This work that has been done proves conclusively that it does increase the comfort and shows to what degree the comfort is increased.

There are many other combinations for which the data will show what effect any change in atmospheric conditions will have on the comfort. Take this practical example. Suppose that you have 90 deg. dry bulb, and say 30 per cent relative humidity, we read on this chart 67 deg. wet bulb. You find it is on the effective temperature line. Now if that air is saturated the wet bulb temperature does not change. Would it increase or decrease comfort? We have changed the effective temperature from 76 deg. to 66 deg., showing very conclusively that the addition of moisture to air does increase the comfort if no other change is made, that is, no other additional heat is added. That is why in industrial work we can cool buildings where temperatures are high by moistening the air. We increase the comfort without refrigeration or actually removing any heat.

DR. E. V. HILL: I believe the curves that Mr. Houghten's tests have determined are not equal comfort lines at all. I think they are

lines of immediate temperature perception. These lines will have to be studied more than they have been up to the present time and I think when the comfort curve—and by the way there is only one—when finally arrived at, will be much closer to the wet bulb than the equal comfort lines shown today.

J. A. DONNELLY: I would like to say something about the relation of equal comfort lines where we have a certain amount of radiant heat in the room. If both test rooms were heated entirely with indirect hot air, circulated air, the data might not hold at all true when there is a certain portion of direct radiation in a room. Personally, I am a great believer in a reasonable relation between radiant and convected heat in a room for occupancy. I think the most terrible thing is a room which is heated entirely with radiant heat and I think and have said there is a disagreeable effect about heating a room entirely with circulated air. The effect of radiated heat on comfort should also be determined.

W. H. CARRIER: The wet bulb evaporation of the body disposes of about two-fifths of the heat and the radiation of about three-fifths at ordinary temperatures. This will indicate to some degree the relative effect of evaporation and dry bulb temperature difference in cooling.

J. D. SMALL: I would like to ask if it is logical to use those equal comfort lines in an ordinarily ventilated room if they were determined in still air?

F. C. HOUGHTEN: The data were determined for still air only. We do not yet know what effect moving air will have, but will determine that later.

W. G. R. BRAEMER: In what way will these effective temperatures affect our ventilating standards or our present code? What changes and additions should be made to our present code of standards for future work along the ventilation line? Are we ready to consider that particular point? It is of very great importance because in different parts of the country laws have been drafted and we have nothing correct to go by.

F. PAUL ANDERSON: It seems to me that this Society should have something like a ventilation committee or commission that would formulate a code. It would be impossible for any one man or scientist to draw conclusions for a piece of work of that sort.

THE PRESIDENT: Our Council has moved that a code committee, of the various different branches, be organized for the very distinct purpose of drafting a heating and ventilating code which will go to counties, cities and otherwise to accept as a ventilating law.

No. 656

NEW DATA ON AIR DUST DETERMINATIONS

BY MARGARET INGELS[1], PITTSBURGH, PA.

Member

A SIMPLE and accurate means for determining the amount of dust in the air is needed. The present methods of measuring dust involve an elaborate laboratory analysis, the results are the averages of a large number of widely varying figures and the operator must be a scientist. An apparatus which can be conveniently used in field work, will give accurate results, and can be easily operated, has been planned by Dean F. Paul Anderson, of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS' Research Laboratory and O. W. Armspach.

A filter through which air is passing may be fancied as a large number of very small orifices, or a smaller number of large orifices, or a number of short-length minute tubes. There will be a drop in pressure of the air across the filter due to the restricted area caused by the filter. The drop in pressure, which is a measure of the resistance to the air flow, will depend on the velocity of the air through the free area of the filter. For a constant air volume and a constant free area the resistance of the filter will remain constant. Air containing dust when passed through the filter will deposit the dust on the filter. The dust deposited may close up some of the orifices, decreasing the number; it may decrease the area of all the orifices or it may lengthen the minute tubes. The result will be an increase in the resistance of the filter.

There is a relation of the dust collected by a filter and the increase in resistance of the filter. On this principle is built the Anderson and Armspach dust determinator.

The development and analysis of the apparatus can be divided into four parts.

1. Design for field use.
2. Standard filter medium.
3. Temperature and humidity effects.
4. Relation of resistance increase to dust collected.

[1] Research Engr., A. S. H.-V. E. Research Laboratory, U. S. Bureau of Mines.

Paper presented at the Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Washington, D. C., January, 1923.

DESIGN FOR FIELD USE

The essential parts, of the dust determinator, set up for the laboratory experiments, can be seen in Fig. 1. The apparatus is operated on the vacuum side of a positive pressure air pump. Air is drawn in at *A*, flows through the filter *B*, through the flow meter nozzle and into the pump. A U-tube pressure gauge measures the drop in pressure in inches of water across the filter. A bypass valve *D* at the pump intake is hand-controlled to give the desired amount of air flow through the filter, measured by the flow meter.

Tests were made to see if the flow meter could be eliminated from the field apparatus as the air volume would drop off as the resistance built up. The bypass valve was set to give a flow of 1 cu. ft. per min. at the

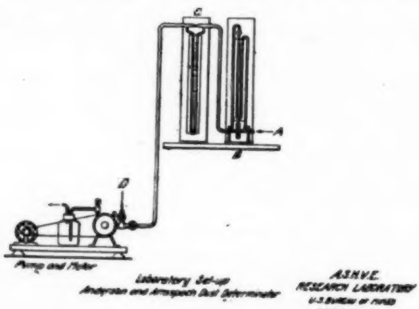


FIG. 1. DUST DETERMINATOR SET-UP IN LABORATORY

start of the test with a clean filter. The volume decreased as the dust was deposited as is shown in the curve, Fig. 2.

The irregularity of the curve shows that a flow meter should be used. The initial resistance of filters vary for equal air volume, so without the flow meter the initial air volume could not be estimated by the initial filter resistance. The motor speed varies changing the pump capacity. With a variable filter resistance, a variable amount of dust in the air, it is advisable not to introduce a third variable of air volume, but the air volume should be held constant.

KIND OF FILTER MEDIUM

A frame made of brass to hold the filter medium is shown in Fig. 3. The gross area of the filter through which the air passes is 0.002376 sq. ft. If 1 cu. ft. of air per min. is drawn through the filter it has a velocity of 421 ft.

CLOTH FILTERS WERE FIRST TESTED

Kind of filter	Threads per sq. in.	Initial Resistance	Velocity
Unbleached Muslin, 1 ply	60 x 65	2.8 in. water @	420 ft.
2 ply		6.4 in. water @	420 ft.

Kind of filter		Threads per sq. in.	Initial Resistance	Velocity
Canton Flannel,	1 ply	45 x 48	5.2 in. water	@ 420 ft.
	2 ply		6.6 in. water	@ 420 ft.
Lonsdale		90 x 96	5.1 in. water	@ 420 ft.
Nainsook		93 x 100	6.1 in. water	@ 420 ft.
Longcloth		78 x 84	6.4 in. water	@ 420 ft.
Linen		51 x 54		@ 420 ft.

Unbleached muslin was used in tests in which two filters were placed in series, each filter consisting of 2-ply of cloth.

The curves in Fig. 4 show that the second filter collected dust. The efficiency of muslin to remove the dust is far below 100 per cent, so the results obtained using it will be only relative values.

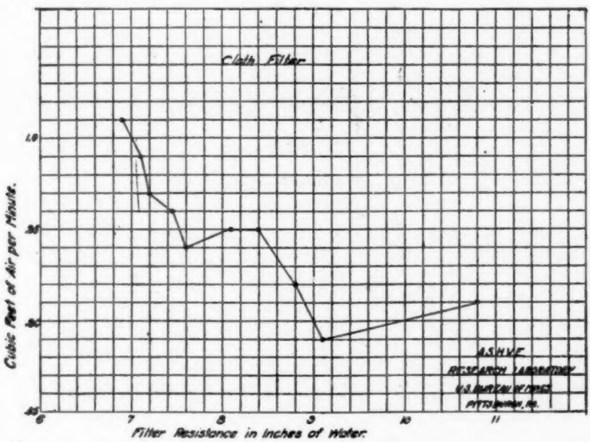


FIG. 2. CURVE SHOWING EFFECT OF DUST DEPOSIT ON AIR VOLUME

A series of comparative tests was conducted during the developments of the dust determinator. Dust was liberated in an air-tight room and simultaneous determinations of the amount of dust in the air were made by the sugar tube, Palmer apparatus, konimeter, thimble filter, new impinger and the dust determinator. A complete report of these tests will be published by S. H. Katz at a later date. Unbleached muslin was the filter used in the dust determinator. Readings taken immediately after the dust had been liberated should indicate a greater amount of dust than readings taken after the dust had had time to settle out of the air.

Curves in Fig. 5 are plotted with the time of readings in minutes against the filter resistance in inches of water. Readings are shown from the time the dust was liberated. Four filters were used, and it is evident that the rate of increase in resistance should decrease as the time lapsed after the liberation of dust. Filters in group *A* should indicate more dust than filters in group *B* and so forth. The five tests were made using unbleached muslin and they all show that muslin does not show this

decrease. This may be due to the large particles dropping first to the level where the samples were taken, and that large particles do not have the clogging effect that smaller particles have, or it may be due to the large particles passing through the openings of the cloth due to their momentum.

Like unbleached muslin all cloth filters showed dust collected on the fourth ply of material through which the air passed. As cloth filters are low in efficiency they will not give absolute results. As their efficiency drops off with high concentrations of dust they will not give reliable results.

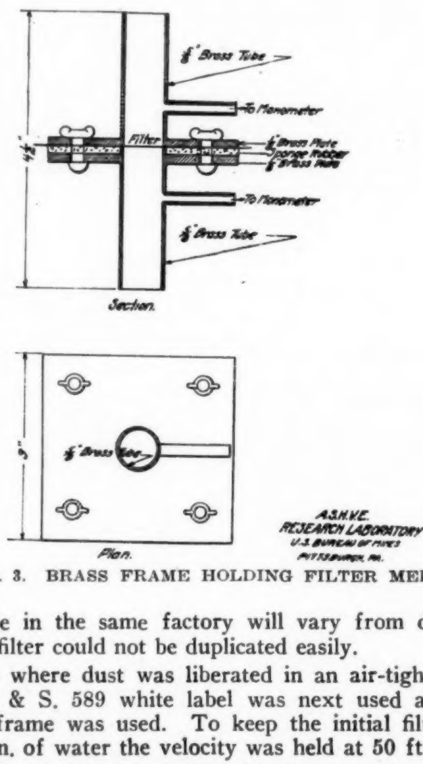


FIG. 3. BRASS FRAME HOLDING FILTER MEDIUM

Cloths made in the same factory will vary from day to day so a standard cloth filter could not be duplicated easily.

In the tests where dust was liberated in an air-tight room chemical filter paper S. & S. 589 white label was next used as a filter. The original filter frame was used. To keep the initial filter resistance as low as 6 or 8 in. of water the velocity was held at 50 ft. per min.

Curves in Fig. 6 show the results with filter paper as a medium, 0.119 cu. ft. per min. circulated: Consistent results were obtained.

A new filter frame to give a gross filter area sufficient to allow 1 cu. ft. per min. of air to be drawn through the paper at 46 ft. linear velocity is shown in Fig. 7.

The gradual increase in area prevents eddy currents and gives uniform distribution across the filter.

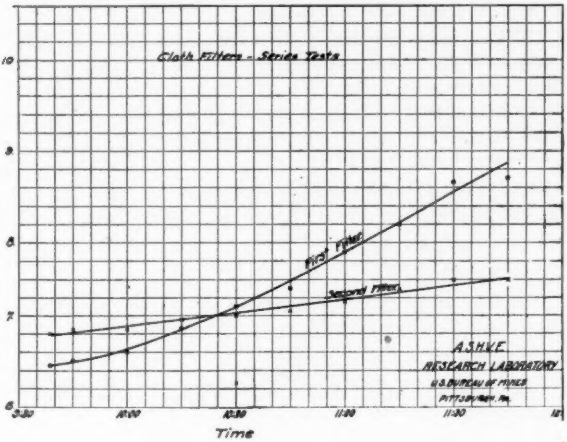


FIG. 4. CURVES SHOWING EFFECTIVENESS OF FILTERS IN SERIES

The dust determinator with chemical filter paper and revised frame was used in the rest of the parallel tests with the other methods of determining dust.

In Fig. 8 are the curves for three tests in which iron dust was liberated. In Test No. 1 there was a high concentration of dust and this is indicated by the curves showing a great increase in resistance over a short time. In Tests Nos. 2 and 3 there were low concentrations, and the slope of these curves is much less. As the time lapsed after the liberation of

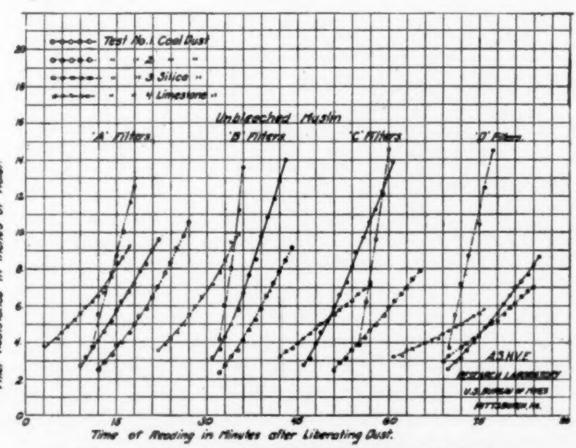


FIG. 5. TEST RESULTS OF FOUR FILTERS

dust the slope of the curves for the filter in each test drops off indicating the dustiness of the air was decreasing.

Curves for two tests in which silica dust was liberated will be noted in Fig. 9, and in Fig. 10 are the curves for two tests in which coal dust was liberated. In each test filter B indicates less dust than filter A, C less than B and D less than C. Fig. 11 shows the filters, which were used in the coal dust tests. The decrease in dust indicated by the decrease in the slope of the curves in Fig. 10 is easily seen from the actual filters.

A scale of degrees of dustiness may be made for each kind of dust and used for optical comparison when determinations of dust are to be made.

CHEMICAL FILTER MEDIUM

As the chemical paper is to be used for the standard filter medium an extensive study of its characteristics was made.

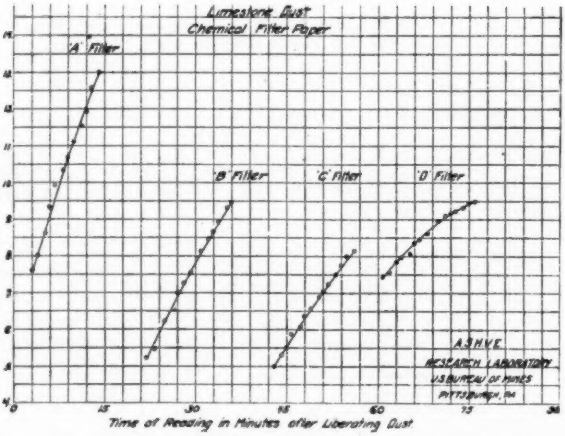


FIG. 6. TEST RESULTS USING PAPER AS FILTER MEDIUM

Paper filters placed in series show no resistance, increase in the second filter, indicating that 1-ply of filter paper is practically 100 per cent efficient on collecting dust.

A series of tests were made to study the resistance-velocity characteristics of filter paper with air containing no dust. Fig. 1 shows the arrangement of apparatus for these tests but there was added at the air intake 3-ply of filter paper to assure clean air at the test filter.

As a check, the velocities were stepped up to maximum and then stepped back to minimum. Room or dusty air was then drawn through the filter for one hour at the rate of fifty feet per minute.

Again clean air was drawn through at the various velocities to see if the characteristics were the same when the resistance of the filter paper had been increased by a dust deposit.

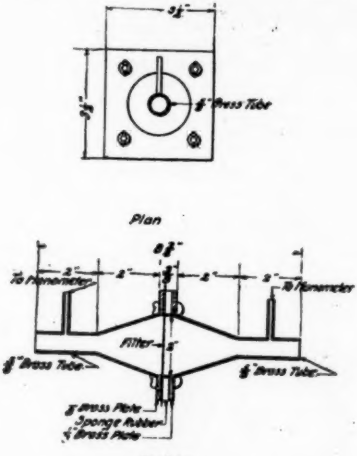


FIG. 7. REVISED STYLE OF FILTER

Curves in Fig. 12 show the data of these tests. It is interesting to note that the velocity-resistance curves have the characteristics of air flow through small orifices, and the dust collected has the effect of decreasing the sizes of the orifices. Between the velocities of 10 ft. per min. and 70 ft. per min. the curve is so nearly a straight line that it can be used as such. If the standard air velocity for dust determinations with this instrument is 50 ft. per min. and it is desirable to make a determination at another velocity, the data obtained can be corrected to 50 ft. for comparisons.

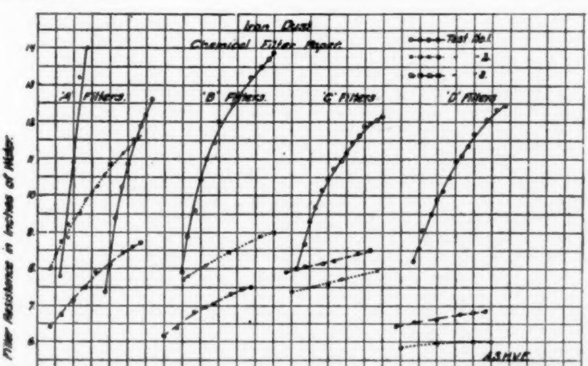


FIG. 8. RESULTS OF THREE TESTS USING IRON DUST IN DIFFERENT CONCENTRATIONS

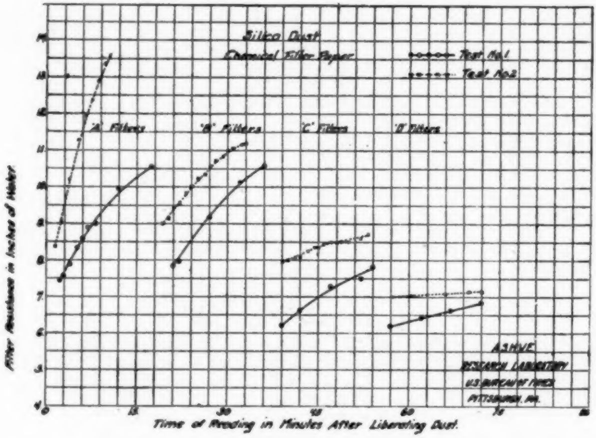


FIG. 9. RESULTS OF TWO TESTS USING SILICA DUST

A series of tests were then made to see if simultaneous determinations would indicate the same air dustiness and the arrangement of apparatus is shown in Fig. 13.

Equal velocity was maintained through the right and left filter. Clean air was first drawn through each filter at 20-40 and 60 ft. velocity, and the resistance-velocity curves plotted. At 40 ft. velocity, room air was drawn through the filter having the smaller resistance at this velocity, and dust was deposited until the resistance of the two filters were equal. Room air was then drawn through both filters simultaneously. Curves Fig. 14 are the results of the simultaneous tests. The resistance-velocity curves

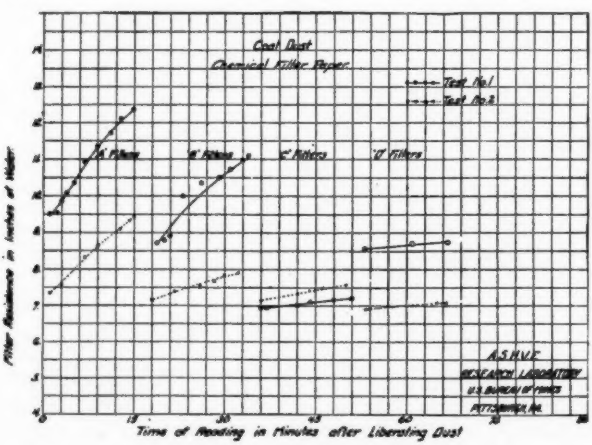


FIG. 10. RESULTS OF TWO TESTS USING COAL DUST

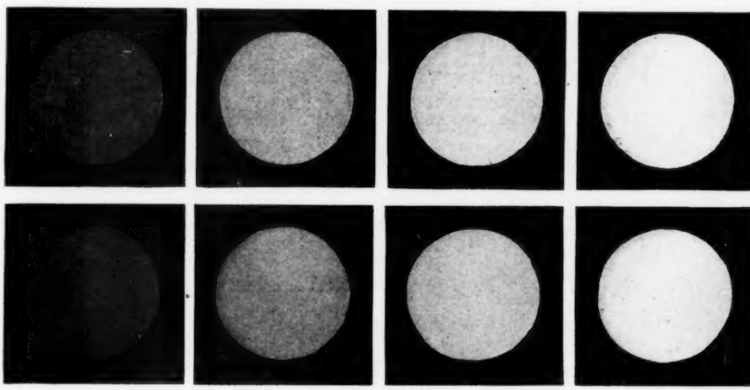
are a repetition of the curves in Fig. 9. In Test No. 1, the left filter was low, but when it was increased to equal the right filter and simultaneous determinations were made with both filters, the rate of resistance increase of both left and right filters follow the same resistance-time curve.

At the end of the tests when clean air was again used and resistance-velocity curves were obtained for the filters with a duct deposit, the right and left filters followed the same resistance-velocity curve. Test No. 2 shows the same characteristics.

In Fig. 15 are shown the filters used in the parallel tests. In each test the final resistance of the right filter equalled the final resistance of the

Test 1

A B C D



A B C D

Test 2

FIG. 11. FILTERS IN COAL DUST TESTS

left filter. The color of the filters would indicate this also. Test No. 1, the final resistance was 15.4 in., in Test No. 2 it was 9.9 in. This difference is shown in the difference of shade of the filters of the two tests.

These tests show that a standard initial resistance at a standard velocity can be established.

FURTHER STUDY OF FILTER MEDIUMS

The chemical filter papers used up to this time were S. & S. 589 white label. A new supply of this paper was different from that used in these tests, the new being made after the war. It is a German made paper. As uniformity in the paper is important, a study of many grades of papers was made.

The table in Fig. 16 shows the papers tested and their resistances to air flow at a velocity of 46 ft. per min.

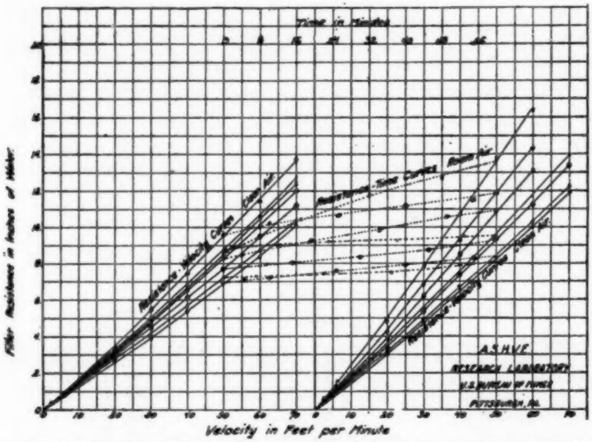


FIG. 12. CURVES SHOWING RESULTS OF TESTS OF CLEAN AND DUSTY AIR

For the manometer tube to be handled easily, its length should not be more than 12 or 14 in. The initial resistance should be sufficiently low to allow for an increase of several inches when working with air of high dust concentrations.

Whatman filter paper No. 41 is uniform, has a low initial resistance, and can be duplicated. This paper was used for the final tests.

EFFECTS OF TEMPERATURE AND HUMIDITY

An apparatus as shown in Fig. 17 was used to determine the effect of humidity on filter resistance. The air was drawn through sulphuric acid

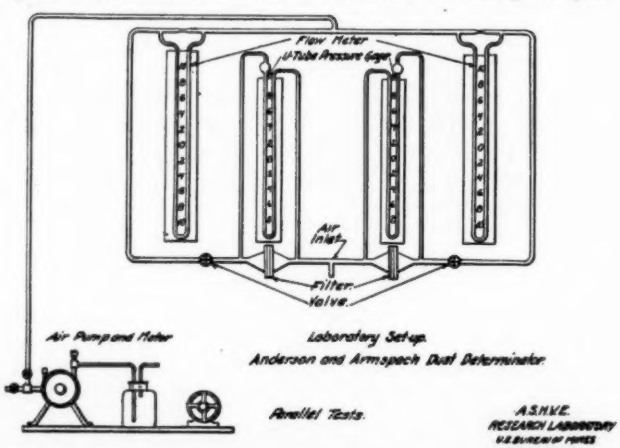


FIG. 13. TESTING APPARATUS SET-UP FOR PARALLEL TESTS

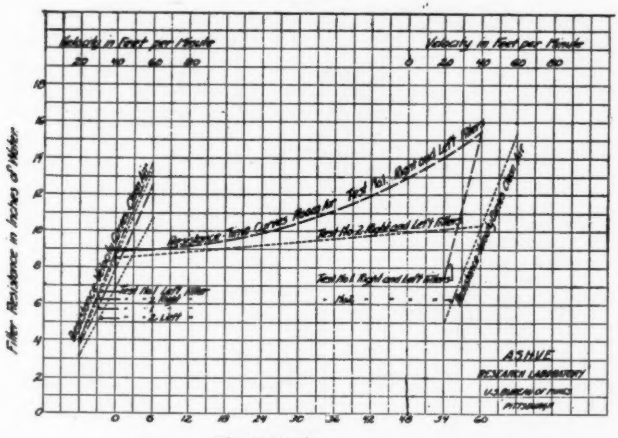
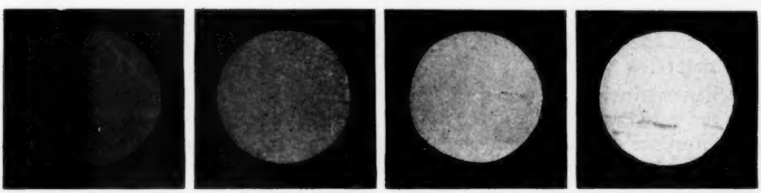


FIG. 14. CURVES SHOWING RESULTS OF SIMULTANEOUS TESTS



Right Filter Left Filter Right Filter Left Filter
Test 1 Test 2
FIG. 15. FILTERS USED IN PARALLEL TESTS

CHEMICAL FILTER PAPERS

Kind of Paper	Number tested	Initial resistance inches of water	Velocity feet per minute
S. and S. 589 White Label	6	9.7—15	46
S. and S. (before war)	5	3.7— 8.3	"
A. D. Little	4	9.2—10.05	"
Whatman No. 1	3	10.2—11.45	"
2	1	13.7	"
3	1	13.7	"
4	1	10.9	"
5	1	More than 15	"
30	1	More than 15	"
40	3	11.7—	"
41	6	3.2— 3.65	"
42	2	More than 15	"
44	1	More than 15	"

FIG. 16. FILTER PAPERS TESTED AND RESISTANCES TO AIR FLOW

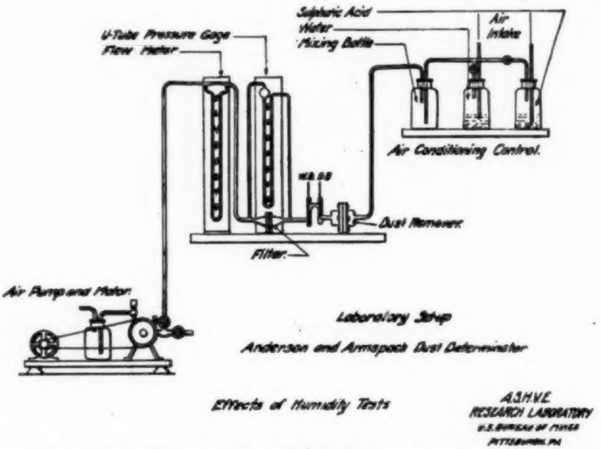


FIG. 17. APPARATUS SET-UP FOR DETERMINING EFFECT OF HUMIDITY ON FILTER RESISTANCE

for low humidities, through water for high humidites or for other humidities, part was taken through each and into the mixing chamber. It was then drawn through 3-ply of filter paper for cleaning and then through the test filter paper. In the line were wet and dry bulb thermometers.

Curves in Fig. 18 show the results of the study of effects of humidity on the resistance of the filter. There is a change in resistance if there is a difference in vapor pressure between the paper and the air. When equilibrium is established, the resistance is low for low humidities and as

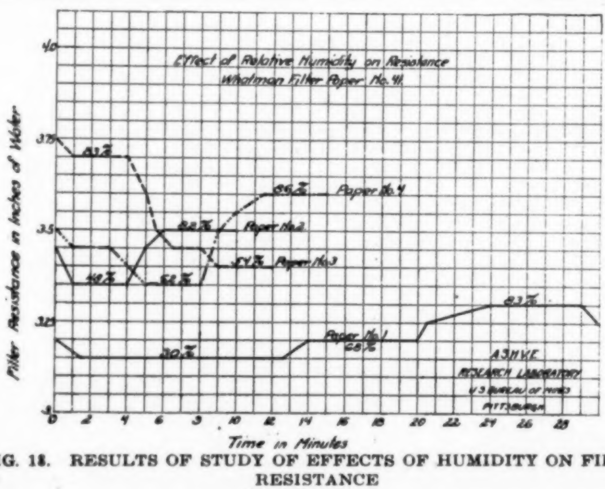


FIG. 18. RESULTS OF STUDY OF EFFECTS OF HUMIDITY ON FILTER RESISTANCE

the per cent humidity increases the resistance increases. The paper picks up moisture decreasing the amount of free area for air to pass through. If the per cent humidity is again decreased the resistance will become lower than it was originally for the same humidity. This is due to the fibre of the paper stretching when in high humidities. The paper has no longer a plane surface but is spherical, increasing the area.

In making dust determinations time must be allowed for equilibrium of the vapor pressures of the paper and air to be established before data are taken. After this equilibrium is established the effect of humidity can be neglected as filter paper is practically one hundred per cent and a change in efficiency due to the dryness or wetness of the filter cannot be detected.

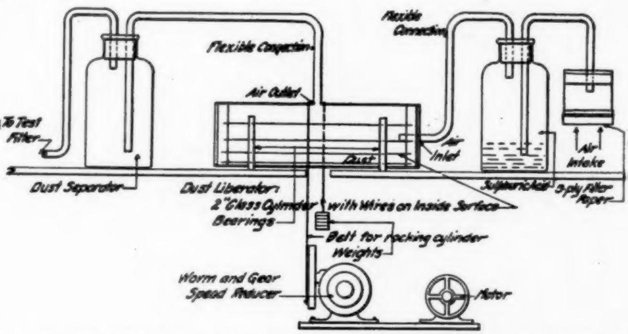


FIG. 19. APPARATUS FOR CONTROLLING THE KIND AND AMOUNT OF DUST IN AIR USED FOR TESTS

RELATION OF RESISTANCE INCREASE TO DUST COLLECTED

The problem of finding the relation of the resistance increase to the amount of dust removed from the air must be solved before the Anderson and Armspach dust determinator will have an absolute value.

At the present time the amount of dust in air is estimated according to the weight of the dust per unit volume and according to the number of dust particles per unit volume.

The weight basis is the better method as there is less likely to be variations in results due to the personality of the operator. The number of particles would be the better basis if the chance of errors was not so great in estimating the number.

The weight basis is the method which will be used to get absolute results with the determinator. If the kind of dust in the air is analyzed and its specific gravity determined, the weight basis can be used with the accuracy in results that the number of particles method would give if the chance of error did not enter into its calculations.

A series of tests were made in which an amount of known dust, liberated uniformly, was added to absolutely clean air, and the increases in resistances for the various weights of dust were determined.

Fig. 19 is a drawing of the apparatus used for liberating the dust. At the intake, the air passes through 3-ply of filter paper for cleaning, then through sulphuric acid for drying. It then enters the cylinder which contains the dust for liberation. The cylinder is rocked on its own axis by the belt connected to the speed reducing motor driven worm and gear. Inside the cylinder are wires held in place by stoppers at each end of the cylinder.

As the cylinder rocks, the dust is carried up on the wires and drops off. The dust is constantly stirred up by this tumbling. The air picks up a large amount of dust and leaves the cylinder and is drawn into the bottle which acts as a separator. By the decrease in velocity and the impact against the bottom of the separator the large dust particles are settled out of the air.

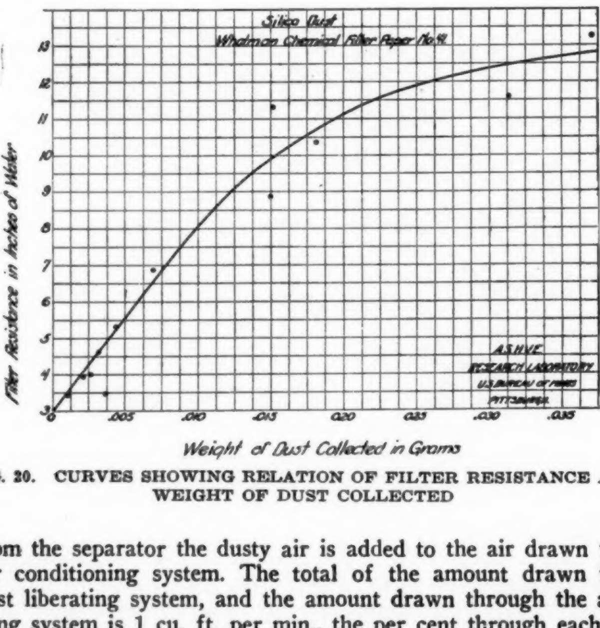


FIG. 20. CURVES SHOWING RELATION OF FILTER RESISTANCE AND WEIGHT OF DUST COLLECTED

From the separator the dusty air is added to the air drawn through the air conditioning system. The total of the amount drawn through the dust liberating system, and the amount drawn through the air conditioning system is 1 cu. ft. per min., the per cent through each system depends on the dustiness of the air desired at the test filter.

Dust from the vacuum cleaner used at the Bureau of Mines was ground, sifted through a 150 mesh sieve and used in the first series of tests.

Filter papers were dried at 105 deg. cent. for 24 hours and weighed. Their clean dry weight varies from 0.3100 to 0.3600 grams. After the dust was collected on them they were again dried and weighed under the same conditions. The difference in weight of the clean dry paper and

the dusty dry paper was used as the weight of the dust collected for the resistance increase of that paper.

Before the vacuum cleaner dust was used it was heated at drying conditions. If there was any oxidation at this temperature it occurred during this preliminary drying, and would not affect the test data.

In Fig. 20 is shown the curve with the filter resistance plotted against the weight of dust collected. The average initial resistance of 3 in. was used as the clean filter resistance.

Silica dust was used in the next series of tests. Air with different concentrations of dust was passed through the filter paper, and at the end of the test the papers containing the dust were fired for 6 hours. The filter paper is ashless. The deposit left in the crucible was used as the weight of dust collected on the paper.

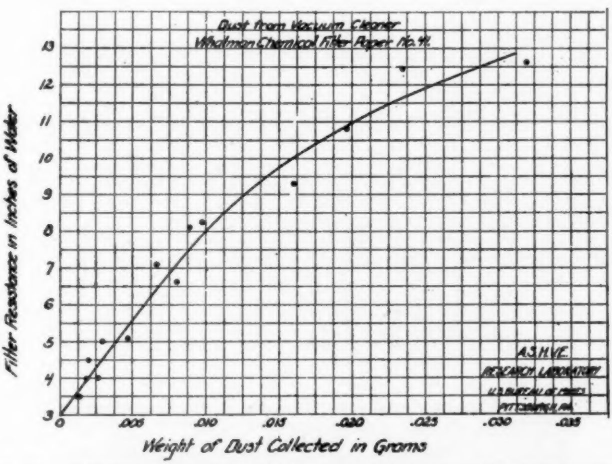


FIG. 21. CURVE SHOWING RESISTANCE PLOTTED AGAINST WEIGHT OF DUST COLLECTED

In Fig. 21 is shown the curve with the resistance plotted against weight of dust collected. The curve for vacuum cleaner dust and silica dust are practically the same.

In Fig. 22 is shown curves of tests made on Laboratory air. The filter resistance is plotted against time. As the rate of air flow was 1 cu. ft. per min. the scale can be read as cubic feet.

These tests were made in a room of about 20,000 cu. ft. content, no forced circulation and with only one occupant. The amount of dust per cubic foot of air over a short period of time may be assumed to be constant.

The curve taken December 27, 1922, indicates a great amount of dust. It follows a straight line up to 8 in. resistance, then curves similarly to the test curves in which vacuum cleaner dust and silica dusts were used.

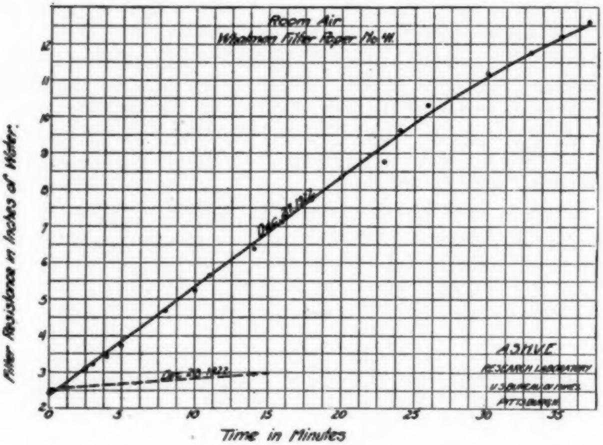


FIG. 22. RESULTS OF TESTS OF LABORATORY AIR

The dust actually occurring in the air may be lighter in weight for the same number of particles than the test dusts. This may be the reason the test dusts show a more decided curve at about 8 in. resistance.

The test taken December 28, 1922, indicates very clean air. The test of December 27, 1922, indicates as much in 1.5 cu. ft. of air as was in 15 cu. ft. of air on December 28. That is December 28 was ten times as dusty.

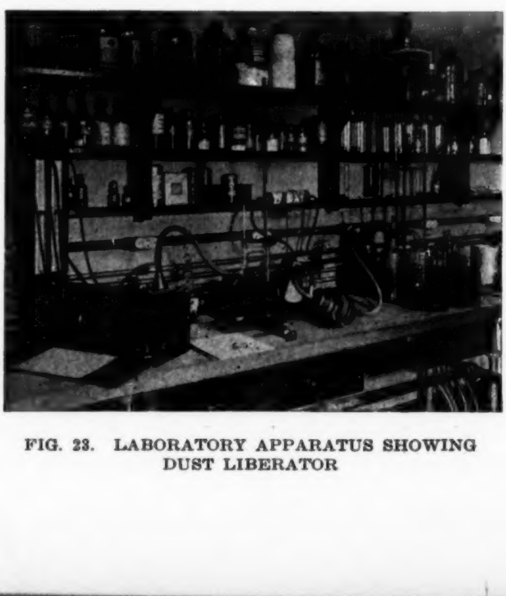


FIG. 23. LABORATORY APPARATUS SHOWING DUST LIBERATOR

The weather bureau gives the following report:

	Barometric Pressure	Rainfall	Wind Velocity
December 27	28.9	0.17 in.	4 miles N.E.
December 28	28.57	0.52 in.	12 miles N.E.

More experiments may develop a relation of these factors with the amount of dust in the air.

For each kind of dust this indicator is to measure, there should be a careful calibration made if absolute values for that dust is to be determined.

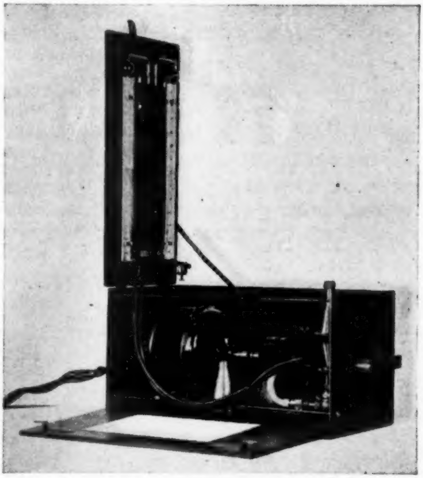


FIG. 24. PORTABLE ANDERSON AND ARMSPACH DUST DETERMINATOR

A test using dry weights of the paper before and after the dust is collected should be made and the weight of dust for the resistance increase determined for that dust.

The dust in a steel mill city may show heavier for the same clogging effect than the dust in a cigarette factory city.

THE PORTABLE APPARATUS

The apparatus in the present design weighs about 40 lb. The fan is direct connected to a 1/6 hp. motor, 3500 rpm. The two manometer gauges measure the air quantity and resistance of the filter. An auxiliary attachment permits absolute clean air to pass through the filter to obtain equilibrium of vapor pressures before starting the tests.

Acknowledgment is made for the valuable suggestions and co-operation of members of this Laboratory, also S. H. Katz and G. W. Smith of the U. S. Bureau of Mines.

DISCUSSION

H. P. GANT: In the dust counter and dust determinator machine what is the assumed difference in efficiency of the filter medium or what percentage of dust that passes into the filter chamber is retained by the filter medium?

MARGARET INGELS: The Bureau of Mines ran a series of tests for the purpose with different mediums and the filter paper will run about 97 per cent efficient in the first two or three seconds of air flow. The dust collecting will increase the efficiency of that paper so it will approach 100 per cent.

No. 657

TESTS OF A MAGAZINE FEED BOILER, WITH SPECIAL METHOD OF SUPPLYING SECONDARY AIR

By John Blizard[1], Pittsburgh, Pa. (Member)

and

J. Neil[2], and A. Pincus[3], Pittsburgh, Pa. (Non-Members)

STEAMING tests were carried out by the fuel section of the U. S. Bureau of Mines, at the request of the American Society of Heating and Ventilating Engineers, on a magazine feed boiler provided with a special method of supplying secondary air, to determine its thermal efficiency and other important factors when fired with various fuels.

The boiler is made by the Reck Heating Co. of Copenhagen, Denmark, and is composed of five principal cast-iron sections. The fuel is charged into a magazine, from which it descends by gravity onto inclined water-cooled grate-bars.

Two sectional views of the boiler are shown in Fig. 1. The fuel is charged through the tightly fitting door *c* into the magazine *b*, from which it descends onto the grate. The water-cooled grate *f* is composed of separate sections like the boiler itself, and the primary air supply to it is admitted past the damper *m* in the ash-pit door. A secondary supply of air is admitted by opening a damper *g* at the back of the boiler, passes through a channel *a* at the edge of the boiler section over the combustion chamber *d*, and enters it through a series openings in the channel. These air openings are shown well in Fig. 2, one of the central sections. On each side of each of these sections are 10 circular air openings and three slots penetrating the outer wall of the air channel. The two outer sections are provided also with an air channel with a series of circular openings.

A door *k* (see Fig. 1) is provided in front of the boiler for examining the fire and removing clinker from the above bars.

The gases from the fuel-bed meet the small streams of secondary air from the air channel, pass round the brick arch *n*, then up and down through the flues *e* to the main flue *i* and the breeching. The door *h*

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on top of the sections is provided for cleaning the flues *e*, and two doors on each end section are for cleaning the main flue *i*.

The water circulates down through two rear ports in the sections and up through two ports near the front. A rib, shown in Fig 1 by broken lines, is situated near the bottom of each section to cause the water to

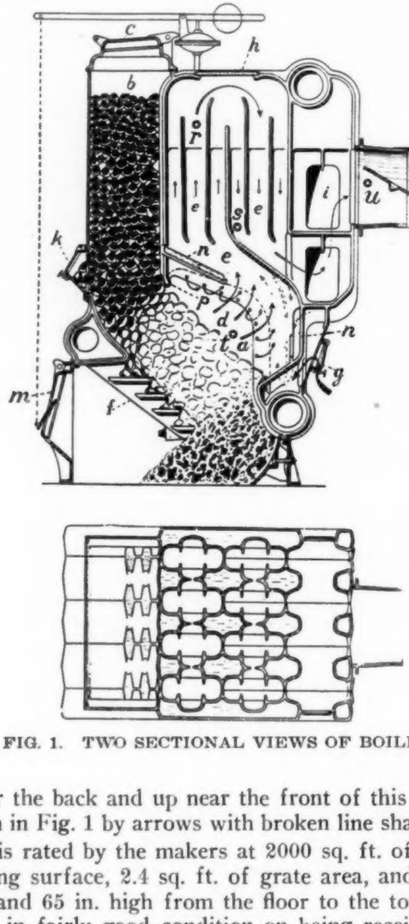

FIG. 1. TWO SECTIONAL VIEWS OF BOILER

pass down near the back and up near the front of this part of the water space, as shown in Fig. 1 by arrows with broken line shafts.

The boiler is rated by the makers at 2000 sq. ft. of radiation, has 66 sq. ft. of heating surface, 2.4 sq. ft. of grate area, and is 28⅜ in. wide, 38⅝ in. long, and 65 in. high from the floor to the top of the magazine cover. It was in fairly good condition on being received, and the flues were cleaned before each test. After running 10 tests it was taken apart, thoroughly cleaned inside and out, and horizontal baffle-plates were placed in the rear flue to lengthen the travel of the gases by causing the gases from the two central flues to pass outward in the last pass, before reaching the breeching. After this change, a test previously run was

repeated at a somewhat lower steam pressure; when the flue-gas temperature was about 70 deg. fahr. lower and the efficiency slightly higher than on a previous test at the same rating. This improvement was attributed to the cleaning, addition of bafflles, careful re-assembly so as to avoid short-circuiting of the gases in the flues, and to the lower steam pressure carried.

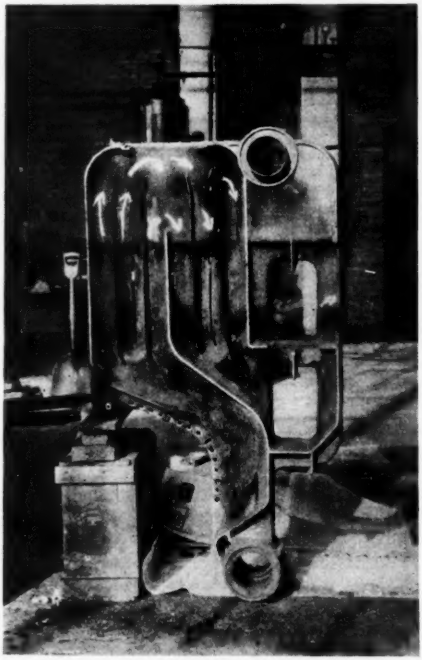

FIG. 2. CENTRAL SECTION OF BOILER

During the first 10 tests P. H. Berggreen, and, during the remainder of the tests, W. E. Reck represented the manufacturers and supervised the operation of the boiler.

Each test was begun and stopped after cleaning the fire and with no fuel in the magazine, so that the volume of fuel on the grate might be estimated by means of gauging, and the condition of the fuel-bed observed through the door *k* in front of the boiler.

The first series of 10 tests was run with a steam pressure of from 3 to 5 lb. per sq. in., and the steam discharged into the atmosphere through an orifice at a constant rate; but in the remaining tests the steam was generated at atmospheric pressure.

The tests were carried out in a thorough manner, and the usual temperature, pressures, etc., were recorded every 30 minutes. Special tem-

TABLE 1. SUMMARY OF RESULTS OF TESTS

F u e l		C o k e				
General particulars						
Number of test		1	2	3	3a	8
Date		May 2-6 1921	May 9-11 1921	May 18-19 1921	May 20-22 1921	Aug. 2-5 1921
Duration	hours	97.65	55.67	43.00	42.02	28.38
Average interval between charging magazine	"	4.4	2.2	1.7	2.0	1.5
Average interval between poking fire	"	3.0	2.4	1.6	2.0	1.2
FUEL						
Ultimate analysis as fired						
Hydrogen	per cent	1.11	1.10	1.11	1.22	1.04
Carbon	"	83.40	81.99	81.89	81.80	84.12
Nitrogen	"	1.03	1.01	1.01	1.01	1.26
Oxygen	"	1.50	1.57	1.63	2.48	1.99
Sulphur	"	0.95	0.83	0.77	0.90	1.02
Ash	"	12.01	13.50	13.59	12.59	10.57
Proximate analysis as fired						
Moisture	per cent	0.73	0.82	0.89	1.85	1.33
Volatile matter	"	3.74	1.82	1.50	3.53	3.50
Fixed carbon	"	83.52	83.86	84.02	82.03	84.60
Ash	"	12.01	13.50	13.59	12.59	10.57
Fuel ratio, F.C./V.M.	"	22.33	46.06	56.00	23.24	24.17
Calorific value of fuel per lb.						
As fired	B.t.u.	12596	12239	12236	12263	12628
Dry	"	12689	12340	12346	12494	12798
Moisture and ash free	"	14436	14284	14308	14333	14333
Evaporative power, heat transferred to water per lb. fuel as fired	B.t.u.	8860	8950	9060	9310	8450
Total fuel fired during test	lb.	1894	1900	2078	2000	2000
ASH AND REFUSE						
Total ash and refuse removed during test	lb.	269	255	270	296	365
Carbonaceous matter in ash and refuse	per cent	26.9	25.5	21.6	23.3	40.6
Ash and refuse removed, per cent fuel as fired		14.2	13.4	13.0	14.8	18.3
DRAFT						
Draft in ash-pit	inches of water	0.034	0.229	0.016	0.170	0.030
Draft in furnace	" " "	0.065	0.280	0.090	0.240	0.170
Draft in stack	" " "	0.112	0.449	0.360	0.520	0.770
AIR						
Temperature of air entering ash-pit	°F.	72	81	82	87	76
Excess air in flue gases	per cent	58	56	33	39	29
FLUE-GASES						
Analysis of dry flue gases by volume						
Carbon dioxide	per cent	12.5	12.7	15.0	14.5	15.5
Oxygen	"	7.8	7.6	5.2	5.9	4.8
Carbon monoxide	"	0.1	0.2	0.1	0.0	0.2
Nitrogen (by difference)	"	79.6	79.5	79.7	79.5	79.5
Mean temperature of flue gases leaving boiler	°F.	420	564	686	660	757
STEAM AND FEED WATER						
Gage pressure - boiler	lb. per sq.in.	4.6	4.5	3.9	3.8	3.4
Temperature of feed water	°F.	64	63	68	79	79
Moisture in steam leaving boiler	per cent	3.3	2.6	2.5	7.0	19.0
RATES						
Per cent of builder's rating		35	63	90	91	123
Heat transferred to water per hour	1000 B.t.u.	172	306	438	443	597
Heat transferred to water per sq. ft. of heating surface per hour	B.t.u.	2610	4640	6640	6710	9050
Fuel fired per hour	lb.	19.4	34.1	48.3	47.6	70.6
Fuel fired per sq.ft.total grate surface per hour	"	8.2	14.4	20.4	20.1	29.8
HEAT ACCOUNT - PER CENT FUEL AS FIRED						
Heat transferred to water (and thermal efficiency)	per cent	70.3	73.2	74.1	75.9	66.9
Heat carried away by steam in flue-gases	"	1.0	1.0	1.1	1.2	1.0
Heat carried away by dry flue-gases	"	10.8	14.9	15.7	15.5	16.3
Heat lost by carbon monoxide		0.5	1.0	0.4	0.0	0.6
Heat lost by carbonaceous matter from ash-pit and grate	"	4.4	4.1	3.3	4.1	8.5
Undetermined losses	"	13.0	5.8	5.4	3.3	6.5
[illegible]	"	100.0	100.0	100.0	100.0	100.0

OF MAGAZINE FEED BOILER

Anthracite, stove size					Anthracite Chestnut	Anthracite Briquettes	Lignite Unscreened	Lignite char Over 1/8" screen	Pittsburgh Bituminous	Illinois Bituminous	Colorado Bituminous
4	10	5	6	7	16	13	12	15	9	11	14
June 7-11 1921	Nov. 16-18 1921	June 27-29 1921	July 19-21 1921	July 26-27 1921	March 15-17 1922	Jan. 30 Feb. 1 1922	Jan. 10-14 1922	Feb. 27 Mar. 1 1922	Sept. 6-7 1921	Dec. 14-16 1921	Feb. 15 1922
119.80	60.58	56.50	37.52	25.42	67.00	64.25	105.00	55.25	25.75	60.00	8.00
8.0	6.7	4.0	3.1	2.6	11.4	9.5	11.7	7.9	2.2	8.0	8.0
3.2	8.8	2.4	2.1	1.2	5.7	4.9	5.0	5.5	0.9	7.3	8.0
2.80	2.99	2.96	2.83	2.89	2.50	3.60	3.20	3.30	5.35	5.15	5.10
78.66	74.84	78.49	77.82	78.57	76.50	81.10	63.40	64.50	73.18	67.23	56.00
0.84	0.77	0.84	0.85	0.84	0.80	1.00	0.80	0.70	1.50	1.37	1.20
4.36	5.44	5.61	4.70	5.13	6.10	4.50	18.40	18.60	8.79	15.97	20.70
0.70	0.76	0.85	1.18	0.84	0.60	0.80	0.80	0.90	1.68	1.45	0.70
12.64	15.20	11.25	12.64	11.73	13.50	9.20	13.40	12.00	9.52	8.73	16.30
2.43	3.55	3.84	2.82	3.28	5.10	1.50	14.30	15.40	1.95	6.52	8.40
5.53	5.57	5.58	6.20	5.95	3.50	10.30	14.00	13.60	37.39	34.70	33.90
79.40	75.60	79.33	78.34	79.04	77.90	79.00	58.30	59.00	51.14	50.05	41.40
12.64	15.20	11.25	12.64	11.73	13.50	9.20	13.40	12.00	9.52	8.73	16.30
14.36	13.59	14.22	12.64	13.28	22.26	7.67	4.16	4.34	1.37	1.44	1.22
12735	12180	12693	12576	12610	12000	13600	9865	9970	13144	11845	9830
13052	12628	13200	12941	13088	12640	13810	11510	11780	13406	12671	10730
14994	14991	14949	14876	14838	14720	15230	13640	13730	14847	13976	13050
8070	8060	8320	7690	7010	8700	7320	4720	6710	5930	6790	
2000	2082	2000	2000	2000	2102	1851	1975	1945	1558	2000	274
471	472	420	493	615	312	466	479	151	236	272	13.5
51.4	47.1	47.9	51.5	66.1	30.0	66.9	59.0	12.9	29.0	40.8	52.7
23.5	22.7	21.0	24.7	30.8	14.8	25.2	24.3	7.8	15.2	13.6	4.9
0.016	0.100	0.110	0.080	0.090	0.021	0.006	0.018	0.018	0.170	0.010	0.024
0.030		0.170	0.180	0.230					0.250		
0.049	0.310	0.310	0.500	0.780	0.330	0.410	0.360	0.450	0.700	0.150	0.165
80	76	84	85	88	75	75	74	69	79	74	78
54	62	50	47	35	51	80	221	94	35	61	25
12.2	11.8	12.7	12.9	13.8	12.8	10.2	6.3	10.3	11.9	11.2	14.5
7.5	8.1	7.1	6.8	5.5	7.2	9.5	14.5	10.2	5.6	8.1	4.7
0.1	0.2	0.2	0.2	0.3	0.1	0.1	0.0	0.2	1.8	0.1	0.3
80.2	79.9	80.0	80.1	80.4	79.9	80.2	79.2	79.3	80.7	80.6	80.5
365	436	512	681	752	509	435	265	444	830	463	485
3.6	0.0	4.0	3.5	4.4	0.0	0.0	0.0	0.0	4.0	0.0	0.0
76	60	80	83	82	55	57	59	51	78	58	57
4.3	2.5	4.9	5.8	14.4	2.1	2.9	6.0	2.2	5.3	3.9	2.9
28	57	61	85	114	66	42	18	49	80	35	57
135	277	295	410	551	321	204	89	236	389	170	275
2050	4200	4470	6210	8350	4860	3090	1350	3580	5890	2580	4170
16.7	34.4	35.4	53.3	78.7	36.9	27.9	18.8	35.2	65.6	25.0	34.3
7.1	14.5	14.9	22.5	33.2	15.6	11.8	7.9	14.9	27.7	10.5	14.5
63.3	66.2	65.6	61.1	55.6	72.5	55.8	47.8	67.3	45.1	57.3	
2.3	2.7	2.6	2.7	2.8	2.3	2.9	3.3	3.7	5.1	4.8	
7.1	10.1	11.0	14.5	13.4	14.0	11.0	9.8	14.8	17.5	11.1	
0.4	0.9	0.9	0.8	1.0	0.5	0.5	0.0	1.2	2.0	0.5	
13.8	12.7	11.5	14.7	23.4	5.4	18.0	21.1	1.5	4.9	6.7	
13.1	7.4	8.4	6.2	3.8	5.3	13.8	18.0	11.5	20.4	19.6	
100.0	100.0	100.0	100.0	100.0	100.0	100.0	100.0	100.0	100.0	100.0	

peratures were recorded also by exposed thermocouples at the point *t* in the combustion chamber, and at the points *r* and *s* in the flues as in Fig. 1

It was intended originally to run a series of tests at about 30, 60, 90, and 120 per cent of rated capacity, with chestnut size by-product coke, stove size anthracite, and Pittsburgh coal. The tests were run as arranged with coke and anthracite, but only one test was run with Pittsburgh coal, because it was found that this coal did not descend automatically from the magazine to the grate, but had to be poked continually. Other tests were run with Illinois bituminous coal, Colorado bituminous coal, lignite char from North Dakota, chestnut size anthracite coal, and anthracite briquettes.

The lignite char was made in a carbonizer installed at Hebron, North Dakota, and experimented on by the U. S. Bureau of Mines in co-operation tion with the University of North Dakota. For one test the char was fed to the boiler as it was received, and for another test it was passed over a 1/8-in. screen to remove the fines before being charged. The size of the unscreened char was as follows:

LIGNITE CHAR SCREEN TEST

Size of hole in screen, inches......	1/2	3/8	1/4	1/8	1/10	1/16	1/20
Char through holes, per cent......	99.7	95.9	63.5	14.6	8.9	3.5	0.0

The Illinois coal, from Westville, had been stored in an open bin since 1916, and had weathered as shown by the proximate analyses of the coal as originally received in 1916 and as fired in 1921:

PROXIMATE ANALYSES OF ILLINOIS COAL

	Moisture	Volatile Matter	Fixed carbon	Ash
As received, 1916............	15.6	35.6	40.0	8.8
As fired, 1921...............	6.5	34.7	50.0	8.8

The Pittsburgh, Illinois, and Colorado coals were all of nut size, while the anthracite briquettes, from the Lykens Valley, were pillow-shaped, being about 1¼ in. long and 2¼in. x 2 in. in cross section.

The results of the tests appear in Table 1. The full results of the test with Colorado coal are not reported, because not enough coal was available to run a test long enough to determine the thermal efficiency and heat losses. The outstanding results of the tests were as follows:

1. That coke, anthracite, lignite char, one shipment of bituminous coal from Illinois which had been stored exposed to the weather for 6 years, and another shipment of bituminous coal from Colorado, could be burned with little poking or other attention to the fuel on the grate or in the magazine except when cleaning the fire or charging the fuel; and that caking Pittsburgh coal was unsuited to this boiler, since it had to be poked at frequent intervals to break up an arch which formed near the top of the grate. The actual periods elapsing between the time of poking the fire and replenishing the magazine during each test are given in Table 1, where it will be observed that the elapsed times are much shorter during the first 10 tests than in the remainder of the tests. The longer intervals between charging in the later tests was due to the difference between the methods of manipulation advised by the

representatives of the makers of the boiler. In the later tests, the fuel was permitted to burn to a much lower level, and larger quantities of fuel were fired at each charging. The comparatively short intervals between charging during the first series of tests must therefore be ascribed to unnecessarily frequent charging of the magazine.

2. That the method of admitting secondary air supply over the fuel-bed permitted the combustible gases rising from the coke fuel-bed to be burned so that only 29 to 58 per cent excess air remained after combustion, those from anthracite with only 35 to 62 per cent excess air, those from Illinois coal with 61 per cent excess air, and those from Colorado coal with only 23 per cent excess air. With all of these fuels, the carbon monoxide left in the gas represented only about 1 per cent or less of the heat of the fuel. Therefore, for these fuels the methods of admitting secondary air was good. On the other hand, the carbon monoxide remaining in the gas when burning Pittsburgh coal with 35 per cent excess air represented 7 per cent of the heat energy of the coal, and the unscreened lignite char was burned with 221 per cent excess air, which was reduced in the trial with the screened char to 94 per cent.

3. The thermal efficiency with coke was highest when operating at about 90 per cent of the maker's rating; at this rating the efficiency was 76 per cent, and dropped to 67 per cent and 70 per cent when operating at 123 and 35 per cent rating, respectively.

With stove-size anthracite, the highest thermal efficiency was 66 per cent at 57 per cent of rating; the efficiency decreased to 56 per cent at 114 per cent rating, and to 63 per cent at 28 per cent rating. Fig. 3 shows the heat account with stove size anthracite, and Fig. 4 that with coke at the various rates of steaming. It will be observed that the efficiency was higher with coke than with anthracite, principally because the loss due to combustible in the ash was less, but also because the loss due to steam in the flue-gas was smaller. The difficulty of burning this anthracite without removing a comparatively large amount of combustible with the ash has been encountered in other tests carried out by the U. S. Bureau of Mines when firing by hand onto plain grates.

When burning the chestnut size of anthracite, the loss due to combustible in the ash was much less than when burning the larger size; and the efficiency increased to about 73 per cent at 66 per cent rating, in spite of a somewhat higher flue-gas loss at this rating.

Anthracite briquettes gave an efficiency of only 54 per cent at 42 per cent rating, largely owing to the high ashpit losses.

The thermal efficiency was as low as 45 per cent when burning Pittsburgh coal at 80 per cent rating, and only 57 per cent when burning Illinois coal at 35 per cent rating. With Pittsburgh coal, the sum of the radiation and unaccounted-for losses was abnormally high, which was probably caused by hydrogen, hydrocarbon gases, tars and soot leaving with the flue-gases. The loss due to carbon monoxide was very high when burning Pittsburgh coal, but fairly low when burning Illinois coal.

When burning the unscreened lignite char at 18 per cent of the maker's rating, the efficiency was only 48 per cent, owing largely to the ashpit losses; but on passing the char over a ⅛-inch screen and removing the fines, this loss was reduced considerably, and the efficiency at one-half rating was 67 per cent.

The test with the Colorado bituminous coal was too short to determine the efficiency, but the low excess air in the flue-gases and comparatively low flue-gas temperature indicated that the efficiency would have been shown to be high had enough fuel been available to continue the test. It will be observed in the full report of the test that the ash and refuse removed was considerably less than the ash charged with the fuel, which also shows that the trial was too brief to give a reliable figure for the economy of the fuel.

SUPPLEMENTARY TEMPERATURE INVESTIGATION

During the first 10 tests with coke, stove-size anthracite, and Pittsburgh coal, temperatures above the fuel-bed were measured at the point *t*

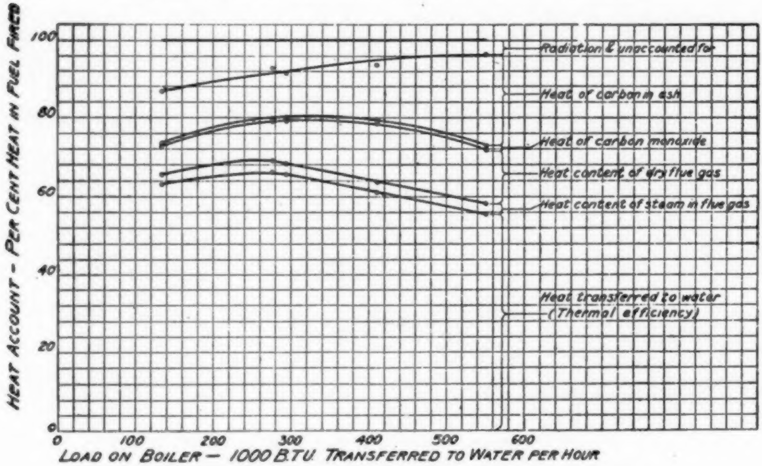

FIG. 3. HEAT ACCOUNT—TESTS WITH ANTHRACITE

(see Fig. 1); and during all the tests temperatures were measured in one of the central flues at two points, one near the top of the first pass at *r*, and another near the bottom of the second pass at the point *s*.

In addition to these temperatures, during the last four tests, temperatures were measured in each of the four flues in the first pass, and during the last seven tests the temperatures in the second pass of both the central flues were measured. The furnace temperatures were measured with exposed platinum and platinum-rhodium thermocouples, those in the first pass with exposed chromel and alumel thermocouples, and those in the second pass with exposed copper and constantan thermocouples.

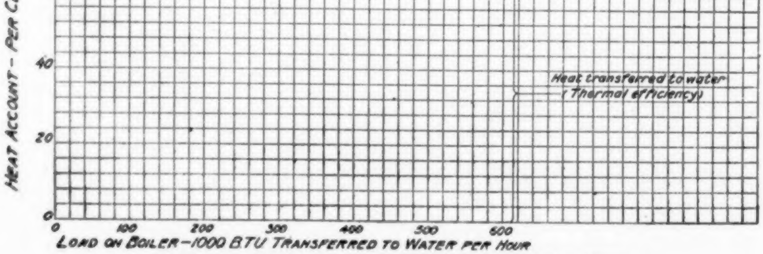

FIG. 4. HEAT ACCOUNT—TESTS WITH COKE

These temperatures are shown graphically at the various rates of steaming in Fig. 5. For stove size anthracite and coke, four pairs of curves are shown—namely, for the furnace, first pass, second pass, and leaving the boiler. Besides the curves, temperatures are shown for the other fuels on ordinates, which are labeled with the names of the fuels burned while the temperatures were observed.

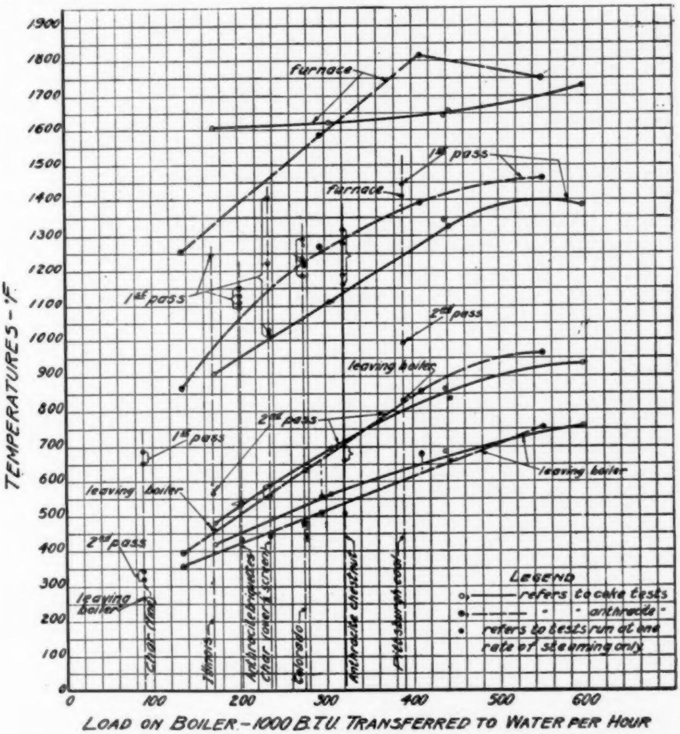

FIG. 5. TEMPERATURES IN FURNACE, FIRST AND SECOND PASSES AND LEAVING BOILER AT REGULAR RATES OF STEAMING

The upper pair of curves show the furnace temperatures, and it will be seen that at the highest rate of steaming and the lowest rate but one the furnace temperatures with coke and anthracite are approximately the same, while at the highest rate but one the temperature was higher with anthracite than with coke by 150 deg. fahr., and at the lowest rating the anthracite furnace temperature was 350 deg. fahr. lower.

At the top of the first pass the temperatures when burning anthracite are higher than when burning coke, and in the second pass and leaving the boiler they are higher at the highest rating and somewhat lower at the lowest rating.

The temperatures in the first pass with the Pittsburgh and Illinois bituminous coals are higher than those with coke and anthracite, which is attributed to the combustible gases, tars, and soot continuing to burn in the first pass. This continued burning accounts for the furnace temperature when burning Pittsburgh coal being somewhat lower than the temperature in the first pass. With the Colorado bituminous coal, however, the mean temperature in the first pass was about the same as with anthracite, so that the gaseous combustible from this fuel evidently burned nearer to the fuel-bed.

Where four points on any ordinate on the chart are bracketed together, they refer to the four temperatures in each of the flues in the first pass. When burning the screened char, these temperatures varied considerably; the temperatures in the two center channels were 1410 deg. fahr., and 1220 deg. fahr., and those in the outer channels 1030 deg. fahr. and 1020 deg. fahr. This indicated that either more gas passed through the central channels or that the gases leaving near the center of the fire were hotter than those near the end section of the boiler. With chestnut size anthracite the temperature in the central flues were also higher than in the end flues. With Colorado coal, the highest temperature was in one of the central flues and the lowest temperature in the other central channel. With anthracite briquettes the temperatures in the four flues were about the same.

The temperatures of the gases leaving the boiler when burning the three bituminous coals show the gases from the Pittsburgh and Illinois coals to be high when compared with those from coke and anthracite, while the gas from the Colorado coal had about the same temperature as that from anthracite at the same rating.

No. 658

EFFICIENCY OF THE PALMER APPARATUS FOR EXPLOSIVE CARBONACEOUS DUSTS

By L. J. Trostel[1], Washington, D. C.

Non-Member

INTRODUCTION

IN selecting a method of dust measuring suitable for testing the heavy suspensions of dust peculiar to grain elevators and mills, the Dust Explosion Laboratory of the Bureau of Chemistry had occasion to test the efficiency of the Palmer dust sampler, among other devices, on a basis of the weight of dust collected, against ten different hazardous dusts.

As the Palmer apparatus finds a wide application in studies of dusty industries, the results here obtained should be of value to everyone interested in the performance of the Palmer dust sampler. No attempt was made to determine the efficiency on a count basis, because this work was done primarily with the idea of determining whether or not a sufficient weight of dust to cause an explosion was present.

The average efficiency by weight, of the apparatus was found to be 98.2 per cent on heavy suspensions, ranging from 284 mg. per cu. m. to 1149 mg. per cu. m. on different explosive dusts of comparatively large particle size.

PREVIOUS INVESTIGATORS

The efficiency of the device on a weight basis has been determined by a number of investigators, but under experimental conditions and with dusts not comparable with the explosive dusts. Palmer[2] showed an average efficiency of 98.3 per cent, using powdered pearl dust. Smyth and Izard[3] pointed out that at a sampling rate of 5 cu. ft. per min., they had found an efficiency of 82 per cent, using a 200-mesh dust with particles of maximum diameter of 40 microns, and that Bills'[4] work done previously

[1] Assistant Dust Explosion Engineer, Bureau of Chemistry.

[2] Methods for Determination of Air Dustiness, Geo. T. Palmer, L. V. Coleman and H. C. Ward. *Amer. Jour. Pub. Health* 6, 1049-75 (1916).

[3] The Practical Hygienic Efficiency of the Palmer Apparatus for Determining Dust in Air, H. F. Smyth and M. S. Izard. *Jour. Ind. Hyg.* 3, 159-67 Sept. 1921).

[4] The Electrostatic Method of Dust Collection as Applied to the Sanitary Analysis of Air, J. P. Bills. *Jour. Ind. Hyg.* 1, 323-42 (1919-20).

would show 81.80 per cent on a 100-mesh rock dust. They reported further that Katz,[5] using air floated silica dust, would have shown 59.85 per cent efficiency and possibly a figure closer to those of the other investigators if he had evaporated his suspensions to dryness to obtain the weight of dust, instead of filtering through paper, whereby, it was presumed, an appreciable weight was lost owing to the fact that the very small particles passed through the filter paper.

These investigators considered only extremely finely divided inert dusts, such as rock and silica dust, whose weight per cubic foot was only a matter of several milligrams. Evidently nothing has been done to determine the performance of the instrument with very heavy suspensions of coarse dusts of varying chemical and physical properties. It further appears that each dust is a separate problem and that the efficiency of the sampler is dependent upon the nature of the dust, its size, the weight used for testing, and the rate of sampling.

DUSTS TESTED

The following dusts were tested: Elevator dust, corn starch, wheat flour, smut spores, and ground oat hulls (explosive hazards in the cereal industries); and ground sulphur, sugar, cocoa, ground hard rubber and wood flour (hazards in the food and chemical industries). All were dried to constant weight and only that portion of the ground oat hulls which passed a 100-mesh screen was used. Except for these differences from the original state, the dusts were tested "as received." Their range of size and character is indicated in the accompanying micro-photographs, Figs. 1 to 10.

According to their appearance, these dusts may be conveniently grouped into three classes:

1. Small, regularly shaped, almost spherical particles, such as starch and smut spores, ranging in size from about 5 to 40 microns.
2. Hairy, fibrous filaments, such as elevator dust and wood flour, ranging in size from 5 microns to 100 times that length.
3. Large, irregularly shaped, amorphous conglomerates, as sulphur and hard rubber, varying in size from 5 to 150 microns.

In order to determine whether or not the difference in specific gravity of these dusts was large enough to explain their behavior in the tests,

TABLE I. SPECIFIC GRAVITY DETERMINATIONS

Elevator Dust	1.52	Ground Sulphur	2.04
Corn Starch	1.47	Sugar	1.59
Wheat Flour	1.47	Cocoa	1.32
Oat Hulls	1.47	Hard Rubber	1.49
Smut Spores	1.34	Wood Flour	1.43

[5] Efficiency of the Palmer Apparatus for Determining Dust in Air, S. H. Katz, E. S. Longfellow and A. C. Fieldner. *Jour. Ind. Hyg.* 2, 167-77 (1920-21).

determinations using a picnometer and cumene were made. The results are shown in Table 1.

EFFICIENCY TESTS

Since it was impossible to set up and maintain a dust suspension of definite concentration for this purpose, the method of introducing a definite weight of dust into the apparatus was made to follow somewhat that used by Palmer[6] in developing the instrument. It consisted in placing the dry dust in a tarred test tube, covering the open end with a piece of No. 9 silk bolting cloth, equivalent to a 100-mesh screen, and inverting the tube over the open end of the glass collecting bowl of the sampler. Gently tapping on the tube produced a fine dust cloud which was easily controlled. Reweighing the tube at the end of each test gave the weight of the dust entering the sampler. In several instances where it was very fibrous, the dust was added to the sampler from a watch glass.

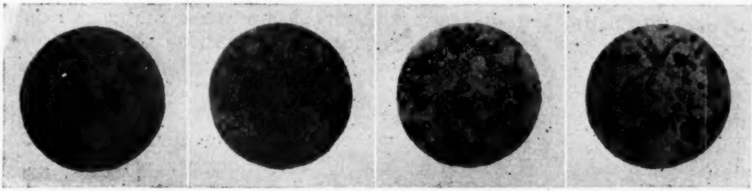

FIG. 1 Elevator Dust — FIG. 2 Corn Starch — FIG. 3 Wheat Flour — FIG. 4 Ground Oat Hulls

Preliminary samples of particularly dusty air from a large export grain elevator, known as a "clean" plant in the trade, were collected with the sampler to find what approximate dustiness might prevail, as a guide in the choice of weights for use in the laboratory efficiency tests. These preliminary results showed a range of 120 to 410 mg. of dust per cu. m., and the conditions prevailing at the time of these observations, according to workmen in the elevator, were about half as dusty as might ever be expected in this plant. In order to determine the performance of the instrument at what would be the maximum condition of dustiness in all plants, from 0.483 to 3.220 grams of dust were added to the sampler in the laboratory efficiency tests, representing a range of dustiness from 284 mg. per cu. m. to 1149 mg. when the sampling period was 20 min., the rate of air flow being 3, 4, and 5 cu. ft. per min.

To determine the weight of dust retained by the sampler, the aqueous suspension from the collecting bowl was transferred to a tarred glass dish and carefully evaporated to dryness, observing the usual precautions for drying such substances as were tested. From 100 to 125 cu. cm. of water proved sufficient to effect a quantitative transference of the dust from the collecting bowl to the evaporating dish, although there could be little objection to the use of larger quantities of water, up to about

[6] Ibid.

200 cu. cm. Frothing occurred, at first, in several instances where the dust contained gluten, but this was finally prevented by adding from three to five drops of caprylic alcohol to the water in the collecting bowl before starting the sampler.

The results of efficiency tests are shown in Table 2.

DISCUSSION

The average efficiency of the sampler, or the percentage of dust recovered from the weight originally added, is 98.2 per cent for all dusts and for all three different speeds of sampling.

Deviation from the average at different sampling speeds in negligible for the weights and types of dust used. The best operating speed is limited only by the type of dust being sampled and the concentration of dust in the air.

The maximum weight of dust which can be handled by the collecting bowl with the greatest efficiency is about 1.5 grams for the fibrous dusts from elevators and 3 grams for the more powdery dusts like corn starch and wheat flour. It is apparent, however, that considerably more than 3 grams of sugar could be handled easily. The best measure of the maximum weight to sample is the effect on the water spray in the collecting bowl, which, of course, must be maintained in a very finely divided condition at all times. The addition of a few cubic centimeters of water as the sampling progresses will help in this respect.

TABLE 2. TABLE OF EFFICIENCIES BASED ON 20 MIN. SAMPLE

Total weight of dust added	Rate at which sampler was operated (cu. ft. per min.) (3) Total volume = 60 cu. ft.			(4) Total volume = 80 cu. ft.			(5) Total volume = 100 cu. ft.			
Grams	0.483	0.966	1.932	0.644	1.288	2.576	0.805	1.610	3.220	Average Efficiency for dust
Milligrams per cubic meter	284	568	1149	284	568	1149	284	568	1149	
Elevator dust	97.7	95.8	96.7	...	96.4	97.0	...	...	98.1	97.0
Corn Starch	97.9	98.4	99.8	98.4	98.9	98.6	98.0	98.8	99.5	98.7
Wheat Flour	98.3	98.6	99.8	99.7	99.4	99.9	99.6	99.9	99.3	99.4
Oat Hulls	97.7	99.8	98.3	...	98.8	99.4	...	98.4	98.3	98.7
Smut Spores	97.7	96.1	99.2	96.3	99.3	99.4	...	99.1	...	98.2
Sulfur	93.8	95.1	96.5	93.2	96.2	97.5	90.4	92.4	96.0	94.6
Sugar	98.6	...	...	99.1	99.4	99.5	99.7	100.0	99.6	99.3
Cocoa	99.4	98.7	97.5	98.6	98.6	...	97.6	97.7	...	98.3
Hard Rubber	98.1	99.6	99.4	99.1	99.1	98.4	98.3	98.8	98.4	98.8
Wood Flour	98.1	98.6	98.3	98.1	98.2	99.0	98.6	99.4	...	98.5
Average efficiency for weight used	97.7	97.9	98.4	97.8	98.4	98.7	97.5	98.3	98.5	
Average efficiency for speed used		98.0			98.4			98.1		98.2*

It has been suggested at various times that the efficiency should increase as a greater weight of dust is sampled. The very large factor of 98.2 per cent obtained in this work, as compared with the general average of about 80 per cent. found by the other investigators when collecting very small weights of dust, bears this out. What appeared to be a further very gradual increase of efficiency of 1.0 per cent was found as the weight of dust added to the sampler was increased from 0.483 gram to

* Grand average for all dusts. Average of last column.

about seven times that amount. This is more apparent from a compilation of the average efficiency for the various weights used in these tests, Table 3.

TABLE 3. AVERAGE EFFICIENCY OF SAMPLER FOR VARIOUS WEIGHTS OF DUST

Weight of Dust Used Grams	Average Efficiency Corresponding Per Cent	No. Determinations Using Weight Indicated
0.483	97.7	10
0.644	97.8	8
0.805	97.5	7
0.966	97.9	9
1.288	98.4	10
1.610	98.3	9
1.932	98.4	9
2.576	98.7	9
3.220	98.5	7

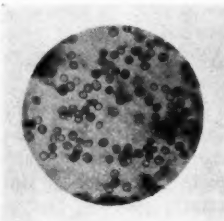

FIG. 5
Smut Spores

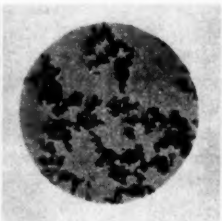

FIG. 6
Ground Sulfur

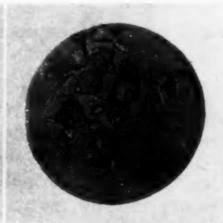

FIG. 7
Sugar

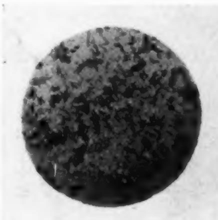

FIG. 8
Cocoa

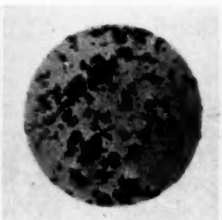

FIG. 9
Hard Rubber

FIG. 10
Wood Flour

The rather large ultimate particle size of these dusts, (5 to 10 microns), is undoubtedly another factor contributing to the high efficiency of the sampler when collecting them. The resistance of these comparatively large particles is so great that they are easily stopped by the water spray and do not tend to pass on with the air stream as is probably the case of the particles of very fine silica and rock dusts, the average diam-

eter of which is from 1 to 2 microns[7] and the ultimate particle size of which is very much less. The effect of size is further brought out in the work of Katz[8] on the efficiency of the Palmer device against tobacco smoke, the particles of which average only about one-quarter of a micron[9]. While Katz's experiments may not be exactly comparable with the work here reported, since he used the Tyndall beam as a basis of measurement instead of the weight per unit volume, yet it is interesting to note that he found an efficiency of 13 per cent against tobacco smoke, which is 2½ times less than the efficiency of 30 per cent found for silica dust tested under the same conditions.

ACKNOWLEDGMENT

This work was done under the general supervision of David J. Price, engineer in charge of development work of the Bureau of Chemistry, to whom the writer is indebted for valuable suggestions. Acknowledgment is also made to B. J. Howard of the Bureau of Chemistry for making the microphotographs accompanying this article, and to H. W. Frevert, also of the Bureau, for several of the analyses performed.

SUMMARY AND CONCLUSIONS

1. The efficiency of the Palmer dust sampler was determined, on a weight basis, on heavy suspensions of ten dusts of varying chemical and physical properties, which are explosive hazards in the elevator, milling, and chemical industries.

2. The sampler retained an average of 98.2 per cent of the dust added to it.

3. There is no evidence to show that the efficiency is appreciably affected by the varying of the sampling rate from 3 to 5 cu. ft. per min. under the conditions of the test.

4. The efficiency appears to increase with the weight of dust collected, up to certain limits.

5. The maximum weight possible to collect efficiently ranges from approximately 1.5 grams to 3.0 grams, depending upon the nature of the dust.

6. The efficiency is higher when dusts of large particle size are being collected.

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[7] Dustiness of Air in Granite Cutting Plants. S. H. Katz and L. J. Trostel, JOURNAL A. S. H. & V. E. (May 1922) 28 : 438.

[8] Efficiency of the Palmer Apparatus for Determining Dust in Air. S. H. Katz, E. S. Longfellow and A. C. Fieldner. *Journal Ind. Hyg.* (1920-21), 2 : 167-77.

[9] An Oscillation Method for Measuring the Size of Ultra-Microscopic Particles. *Journal Amer. Chem. Soc.* (1919), 41 : 312-19.

No. 659

THE SEMI-ANNUAL MEETING

1923

THE largest meeting on record was held by the Society at the Drake Hotel, Chicago, May 21 to 23. With 573 members and guests registered, the business and technical sessions were unusually well attended and the meeting was a great success in every way.

The ladies were kept busy every minute of the time with sight-seeing trips, luncheons, theatre parties, teas, and other social functions, and a dinner dance at Chicago's famous South Shore Country Club was greatly enjoyed by all the members and guests who were able to attend.

It was a meeting of record accomplishment in that the Code for Testing of Fans was unanimously adopted as a standard of the Society, that great progress was made in the development of the Code of Minimum Requirements for the Heating and Ventilation of Buildings, and the new Constitution and By-Laws was formally adopted. The Research Laboratory contributed a good share of valuable data and the papers prepared by members on other subjects of interest were of the highest order.

The Committee on Increase of Membership reported a marked increase in membership, the total for the Society now being 1694. With 118 unaffiliated and 130 under consideration it was expected that the membership would reach 2000 before the end of the year.

Annual reports of Chapters had been received from Colorado, Illinois, Massachusetts, Ohio, Philadelphia, St. Louis, and Western New York. At the suggestion of H. M. Hart these reports were read by title and were to be published in the September JOURNAL.

At the closing session of the meeting the Report of the New York State Commission on Ventilation was introduced and discussed at some length. On motion of Homer Addams, the report was referred to a committee for consideration, and its recommendations were to be presented at the next Annual Meeting.

PROGRAM

FIRST SESSION

Monday, May 21, 9:30 a. m.

Business Session:
Welcome Address
Response by President
Annual Reports of Chapters
Report of Committees:
a. Chapters
b. Increase of Membership
c. Guide Publication Committee
d. To Confer with A. S. M. E. Boiler Code Committee on Heating Boilers
e. Standard Code for Testing Fans
Report on Building Code of U. S. Department of Commerce.

SECOND SESSION

Monday, May 21, 2 p. m.

Professional Session:
Discussion on Code of Minimum Requirements for the Heating and Ventilation of Buildings.

CONTINUED SESSION

Monday, May 21, 8 p. m.

Discussion on Code of Minimum Requirements for the Heating and Ventilation of Buildings.

THIRD SESSION

Tuesday, May 22, 10 a. m.

Research Session:
Report of Committee on Research
Address by Director F. Paul Anderson
Paper: A Field Method for Determination of Ozone, by W. P. Yant, G. W. Jones and F. C. Houghten.
Paper: Ventilation of Garages, by G. W. Jones and S. H. Katz.
Paper: Further Observations in Dust Determination, by Margaret Ingels.
Paper: Heat Transference............from page 208

CONTINUED SESSION

Tuesday, May 22, 2 p. m.

Paper: Heat Transference and Combustion Tests in a Small Domestic Boiler, by John Blizard, W. M. Myler, Jr., J. K. Seabright, and C. P. Yagloglou.
Paper: Further Study of Physiological Reactions, by W. J. McConnell, F. C. Houghten, and F. M. Phillips.
Paper: Determination of the Comfort Zone, by F. C. Houghten and C. P. Yagloglou.

FOURTH SESSION

Wednesday, May 23, 10 a. m.

Heating Session:

Paper: Insulation of Cold Surfaces to Prevent Sweating, by L. L. Barrett.

Paper: Heating with Condenser Circulating Water, by J. E. Williams.

Paper: Registers Temperatures in Warm Air Heating, by V. S. Day.

Paper: District Heating Progress in Boston, by D. S. Boyden.

FIFTH SESSION

Wednesday, May 23, 2 p. m.

Ventilation Session:

Discussion of Report of N. Y. State Commission on Ventilation.

Report on Ventilation Code of State of Wisconsin.

Paper: Heating and Ventilation in Chicago Schools, by John Howatt.

REPORT OF DIRECTOR OF THE RESEARCH LABORATORY

It is not my purpose on this particular occasion to discuss the value of Research as far as the Society is concerned for I believe every member has become a convert to the constructive importance of the Research Laboratory. One of the greatest tributes to industrial research was paid by Floyd W. Parson in a recent article in the *Saturday Evening Post* entitled, "Looking Beyond the Rim," in which it was pointed out that government institutions are not proper places to develop research but only the most fruitful results may be obtained in universities, industries and at the hands of Societies like this.

The Research papers presented today simply show the progress at the Research Laboratory and are primarily to outline the new studies being conducted at the Laboratory. The important work on Ozone by Mr. Yant is perhaps the most valuable single investigation made on this subject. We have been looking at the possibilities of Ozone in Ventilation for a number of years and it appears now that we have every reason to feel that there is virtue in ozone and that it can be used with great profit in ventilation, though actually what its effect is we do not know. Two things that have to be determined if ozone is to be used intelligently, first, whether it is injurious as made by present day generators, and secondly, how to measure the ozone in atmosphere. We will hear about this latter phase of the subject today.

The subject of Dust has interested us greatly at the Laboratory and it is our belief that we have developed a method which will be adopted quite universally for the determination of dust in atmosphere. The material that will be presented today undoubtedly will demonstrate to you just what a delicate instrument we now have available for accurately determining the amount of dust in the atmosphere. It is our hope to have an analysis of commercial dust eliminating apparatus to present at the next Annual Meeting.

The paper on Heat Transference and Combustion in a Small Domestic Boiler illustrates the co-operative work carried on between the Research Laboratory and the U. S. Bureau of Mines, and the facts presented are very valuable.

Two other papers of great interest and importance are those dealing with the Comfort Zone and with Physiological Reactions to Various Temperatures and Humidities.

Our future problem contemplates a presentation of data on the heat meter which has been developed by Percy Nicholls which will measure the flow of B.t.u. through a given area. The heat meter will be presented to the world at the next Annual Meeting.

Another phase of the heating problem which is being studied and results are very conclusive on the flow of steam and steam mixtures through pipes and orifices. A definite plan has been laid out to determine the capacity of pipes when used for dry and wet returns and it is expected that a report on this interesting heating problem will be forthcoming for the next meeting.

At the Laboratory we are interested in every problem in heating and ventilating and I hope that more members of the Society will visit the Laboratory and become acquainted with the work that is being done. I wish to commend the spirit of the Laboratory staff which is a perfect tribute to the loyalty to the problems on which they are engaged. I hope that the results presented today will be gratifying to the members and that they will enter into the free discussion of the subjects and also voice their opinion of the general character of the work that is being carried on in the Laboratory.

As the time is so limited for the presentation of the vast amount of information that the Laboratory has to announce today, I will not even take the time to present to the Society, even by topic, all of the investigations that are being pursued at Pittsburgh. In the future it will be impossible to record all of the laboratory findings at the Annual and Semi-Annual Meetings. It will be necessary to have some results in the JOURNAL from month to month.

We are all very grateful for the hearty and enthusiastic support given to the Laboratory. We will endeavor to present such facts to the scientific world each year that will fully justify the co-operation and good-will shown at every hand.

It is my pleasure to introduce to this eminent body these scholarly scientists who will present the most interesting and valuable papers that have been announced by the official program.

F. PAUL ANDERSON, *Director*.

No. 660

REPORT OF THE NEW YORK STATE COMMISSION ON VENTILATION

NOTE:—The final report of the New York State Commission on Ventilation has just been issued in the form of a book containing some 600 pages.

Through the kind permission of the publishers, E. P. Dutton & Co., New York, we are able to present, below, the general conclusions as formed on pages 199 and 519 to 529 of this report.

This presentation is for the sole purpose of informing our members and readers regarding these general conclusions of this commission and must not be construed as reflecting the full text of the report or any opinionated views as to its correctness or value.

Special attention is called to the following points of interest in connection with this report:

(1) *Many of the findings appear to be at wide variance with established ventilation practice and must needs, therefore, establish new practices or stand refutation as the test of time may determine.*

(2) *Most of the problems to which answers are suggested are as deep as the entire science of heating and ventilating and must needs, therefore, finally gravitate to this Society for a large burden of their necessary proving out.*

(3) *The scientific research work which the Society is now carrying on through its Research Laboratory and other research facilities, on the subject of ventilation, is designed to solve these and many other such problems and will no doubt be of far-reaching value and interest in its bearing upon this report.*

In the meantime it is suggested that a study be made of these conclusions and of the full text of this report in order that we may best save the welfare of the public by assisting in the proper development of this important branch of our science.

GENERAL CONCLUSIONS IN REGARD TO THE EFFECT OF VARIOUS ATMOSPHERIC CONDITIONS UPON HEALTH, COMFORT AND EFFICIENCY[1]

The experiments of the Commission have in general confirmed the conclusion of earlier investigators that the first and foremost condition to be avoided in regulating the atmosphere of occupied rooms is an excessively high temperature. We have found that even slight overheating (24 deg. cent. or 75 deg. fahr.) produces the following harmful results.

1. A burden upon the heat-regulating system of the body, leading to an increased body temperature, an increased heart rate and a marked decrease in general vaso-motor tone as registered by a fall in the Crampton index.

2. A slight but definite increase in rate of respiration.

3. A considerable decrease in the amount of physical work performed under conditions of equal incentive—a decrease amounting to 15 per cent at 24 deg. cent. (75 deg. fahr.) and to 28 per cent at 30 deg. cent. (85 deg. fahr.).

4. A markedly abnormal reaction of the mucous membranes of the nose, leading ultimately to chronic atrophic rhinitis and when followed by chill, producing a moist and distended condition of the membranes calculated to favor bacterial invasion. In animals, exposure to high atmospheric temperatures, particularly when followed by chill, diminishes the protective power of the blood and markedly increases general susceptibility to microbic disease.

For these reasons we believe that the dangers of room overheating are far more serious in their effect upon human health and efficiency than has been generally realized and that every effort should be made to keep the temperature of the schoolroom, the workroom and the living-room at 20 deg. cent. (68 deg. fahr.) or below.

With regard to the problem of relative humidity it is obvious that a high moisture content combined with high temperature must always be harmful, since the effect of a humid atmosphere is to decrease the heat loss from the body by evaporation. The specifically harmful influence of unduly low humidity which has been postulated by various writers upon ventilation has, on the other hand, not been apparent in our investigations.

Our results in regard to the chemical composition of vitiated air (temperature and humidity effects being excluded) have been generally negative. In two respects, however, our experiments suggest that some chemical constituents of the air of an unventilated room may be objectionable. Such air appears (1) to decrease the appetite of human subjects for food and (2) to diminish substantially the amount of physical work performed under conditions of equivalent stimulation.

We may conclude then that the primary condition of good ventilation is the maintenance of a room temperature at 20 deg. cent. (68 deg. fahr.) or below without the production of chilling drafts; but that it is also important on account of certain subtle but real effects of vitiated air upon appetite and inclination to work, to provide for an air change sufficient to avoid a heavy concentration of effluvia such as was associated in our experiments with a carbon dioxide content of 23 to 66 parts per 10,000.

The methods of attaining this latter result, and of maintaining desirable temperatures under the conditions of actual schoolroom life, will be fully discussed in Part II of our report.

GENERAL CONCLUSIONS IN REGARD TO THE RESULTS PRODUCED BY VARIOUS METHODS OF SCHOOLROOM VENTILATION[2]

OBJECTS AND SCOPE OF THE INVESTIGATION REPORTED IN PART II

The previous investigations of the New York State Commission on Ventilation, reported in Part I of this report, have indicated with some clearness the desirable ends to be attained by the ventilation of occupied spaces. These studies show that the effects of chemical impurities in the air of the ordinary occupied room are of relatively slight importance in their influence on health and comfort, and that the main objective of ventilation is the preservation of the proper degree of temperature, humidity, and air movement which will facilitate the elimination of heat from the body surfaces and will stimulate those surfaces by an agreeable degree of coolness and air movement. At the same time our investigations indicate that for esthetic reasons, and also on account of a minor but real influence of the odoriferous constituents of the vitiated air upon appetite, occupied rooms should be kept free from noticeable odors.

We have concentrated our attention chiefly upon six ventilation procedures, which in the body of this report have been described as follows: *Method 1*, window

ventilation alone; *Method 2*, window ventilation with gravity exhaust; *Method 3*, window ventilation with fan exhaust; *Method 8*, plenum supply with gravity exhaust; *Method 9*, plenum supply with fan exhaust; *Method 10*, recirculation.

For the purpose of investigating the effects of the various methods described above we equipped two elaborately planned experimental schoolrooms in School 51, Bronx, New York City, and in the course of the work have studied rooms in about 30 other schools in New York City, Springfield, Mass., Fairfield, Conn., and Minneapolis, Minn. The work has extended more or less continuously over five school semesters including the spring and fall terms of 1915, the spring and fall terms of 1916, and the spring term of 1917. In all, some 216 different schoolrooms were included in the study.

Our judgment as to the results obtained by these various methods of ventilation has been based upon (*a*) the physical conditions maintained as determined by observation of temperature, relative humidity, rate of heat loss as observed by the kata-thermometer and comfortimeter, rate of evaporation as determined by the atmometer, carbon dioxide content, and measurements of air flow (*b*) the sensations of comfort as evidenced by votes of teachers and of the Commission's own observer expressed on various arbitrary scales to register impressions as to temperature, moisture, odor, and general agreeableness of the atmosphere; (*c*) the physical condition of pupils at the beginning and end of experimental periods as determined by detailed physical and medical examination; (*d*) the mental progress of pupils as evidenced by psychological tests and by records of ordinary classroom work; (*e*) the prevalence of respiratory disease as determined by the daily observation of nurses assigned to examine the children in the classroom and to visit absentees in their homes.

THE VENTILATION OF THE SCHOOLROOM BY THE USE OF WINDOWS ALONE

Results obtained in schoolrooms ventilated by windows alone without the use of gravity exhaust ducts have been uniformally unsatisfactory. In our first comprehensive studies conducted during the winter of 1915-1916, rooms ventilated by this method showed a highly variable temperature, were objectionably chilly near the floor, and were characterized by a poor aeration and the presence of offensive odors. A schoolroom provided with direct radiation and ventilated in this way may be kept in a comfortable condition in mild weather if liberal use is made of windows, doors, and transoms. With the onset of colder weather it becomes necessary to cut down the area of these openings to preserve desirable warmth. Under these conditions, the natural flushing which takes place in the milder weather is impeded and this results in an accumulation of the products of human exhalation and a condition of general atmospheric stagnation. The air has a high carbon dioxide content and possesses an unattractive odor. The heat given off from radiators rises to the ceiling where it tends to accumulate instead of diffusing about the room. The floor in consequence becomes colder. Air enters through the window into a blind pocket. The weight of the colder outdoor air cannot act to advantage in forcing out the warmer room air, as there is no convenient outlet. If windows are opened at both bottom and top, a circulation may be induced in the immediate vicinity of the window, the incoming cold air tending to drop to the floor and thus forcing out the heated air through the open window top. This condition is not conducive to the comfort of the occupants. If radiators are located immediately beneath the windows and the windows are fitted with deflectors, the incoming cold air cannot drop to the floor so readily but mixes with the rising heated air and is short-circuited out through the open window top, without effectively sweeping over the room.

These results are only what might be expected in view of the experience of earlier years, and indicate that the demand made a hundred years ago for the provision of some definite means by which air could enter and leave the schoolroom was fully justified.

In view of the fact that the ventilation of the schoolroom through open windows fitted with cloth screens has been widely advocated in certain quarters, it should be mentioned that the Commission has made a special study of this point and

found that rooms ventilated in this way are characterized by particularly inadequate aeration as well as by interference with the lighting of the schoolroom, which conditions in our experience have resulted in grave discomfort and serious complaints.

We are definitely of the opinion that for the average schoolroom which is to be maintained, not as an open-air room, but at a temperature between 65 deg. to 70 deg. fahr. (18.3 deg. to 21.1 deg. cent.) ventilation by windows alone without the use of exhaust ducts is generally likely to prove unsatisfactory.

VENTILATION OF THE SCHOOLROOM BY THE USE OF WINDOW INLETS AND GRAVITY EXHAUST

A very considerable part of the resources at our disposal were devoted to the study of the system of ventilation which involves the admission of air over slanting window boards (with radiators below) and the removal of vitiated air from a gravity exhaust duct on the inner wall. This procedure is well exemplified by the Sherman School at Fairfield, Conn., where it was introduced on the initiative of S. H. Wheeler. The Commission was able to study this school at Fairfield and has also equipped numerous rooms on the same plan in various New York City buildings. During the two school seasons covered by our study this method was almost constantly under observation, and we have extensive data from which to judge of its adequacy.

The records at our disposal furnish ample evidence that this system of ventilation can be used under a wide variety of weather conditions with excellent results, so far as the comfort of the occupants of the schoolroom is concerned. The method avoids the stagnation characteristic of ventilation by windows alone but does not of course accomplish aeration as complete as that characteristic of plenum ventilation. On the other hand, as will be noted below, we have ample evidence that the diminished aeration does not tend to produce discomfort or to injure the health of the occupants. Some difficulty is experienced from back drafts through the exhaust flues in the use of this system, because of which fact shut-off dampers should be installed at the exhaust openings so that they may be partially closed off when such difficulties occur. So far as temperature is concerned the conditions characteristic of this type of ventilation seem to be the most agreeable with which we have dealt in our studies. With the temperature maintained in the neighborhood 67 deg. fahr. (19.4 deg. cent.), a schoolroom of this type produces a very agreeable impression upon the occupants and upon visitors.

CONDITIONS ESSENTIAL TO SUCCESS IN VENTILATION BY THE USE OF WINDOW INLETS AND GRAVITY EXHAUST

The success of the method of schoolroom ventilation by window inlets and gravity exhaust depends upon the fulfilment of certain essentials which may be summarized as follows:

(*a*) Radiators must be located beneath the windows and extend for the full width of the windows from which the air supply is to be derived. These radiators, because much larger than those customarily installed in ordinary plenum systems, should be either automatically controlled by intermediate acting thermostats or equipped with fractional or modulating hand-controlled valves, placed at the top of the supply end of the radiator. Even when automatic control is included it is best to supplement it by the provision of hand control as well; and standard metal radiator shields are desirable to protect the pupils nearest the radiators from excessive heat. It is to be noted that the use of intermediate acting thermostats or modulating hand-controlled valves presupposes the use of a vapor or vacuum steam heating system.

(*b*) Deflecting boards of some satisfactory type should be placed at the bottom of the windows. Devices which include small box-like openings and devices which involve the use of filtering screens of various types are undesirable. A plane glass deflector one foot high is fairly satisfactory, but the

best results may be obtained by the use of curved vane deflectors, which secure the most equable distribution of the air. The windows in the use of this method should open from the bottom and not from the top.

(*c*) In order to avoid certain practical difficulties met with in this study, it is recommended that the windows should be so constructed as to open easily from the bottom, and that window shades should be firmly attached to the window frame, the best arrangement being that which includes two shades anchored midway between top and bottom, one to be pulled upward and the other downward, these shades being so guided by cords and pulleys as to avoid the shaking of the screens by the incoming air flow.

(*d*) Exhaust ducts having a total area of not less than 8 sq. ft. (0.75 sq. m.) for an ordinary schoolroom should be provided on the wall opposite the windows. These exhaust openings should be conveniently dampered so that their area may be adjusted to varying weather conditions. The exhaust ducts should be carried by through the interior of the building so as to avoid chilling and the tendency to back drafts should be further reduced by placing heating coils in the exhaust duct.

(*e*) The schoolroom should not be overcrowded. The successful results reported by us with this method of ventilation have been obtained with a cubic space allowance of 250 cu. ft. (7.1 cu. m.) per 2nd grade child (39 children in an ordinary schoolroom) and with a cubic space allowance of 310 cu. ft. (8.8 cu. m.) per 6th grade child (31 children in an ordinary schoolroom).

(*f*) A large thermometer with 68 deg. fahr. (20 deg. cent.) clearly indicated as a danger point should be displayed in a prominent position on the teacher's desk.

VENTILATION OF THE SCHOOLROOM BY PLENUM SUPPLY AND FAN EXHAUST

In order to avoid the tendency to occasional back drafts manifest in the method of window supply with gravity exhaust it would seem reasonable to consider the supplementing of window ventilation by the use of exhaust ducts provided with suction fans. This method has not been studied by us in any very exhaustive way, but such work as has been done from this point, reported in Chapter XIX, did not yield favorable results. Back drafts, it is true were prevented, but on the other hand air was drawn in from the windows or from the corridor in too large an amount and with too high a velocity so as to produce an uncomfortable chilliness and marked inequalities in temperature. It would be possible to avoid this evil by the use of a suction exhaust of sufficiently small amount, but it should be clearly recognized in such a case that the object of the exhaust is simply to overcome the danger of back drafts and not to maintain a large and continuous air flow such as has generally been contemplated in fan ventilation. The very essence and virtue of window ventilation lie in the avoidance of constant and powerful air currents.

VENTILATION IN THE SCHOOLROOM BY PLENUM SUPPLY AND GRAVITY EXHAUST

The ventilation of the schoolroom by the ordinary standard method of plenum supply and gravity exhaust yields very satisfactory results in a plant properly installed and properly operated. This method gives excellent aeration but maintains a uniform temperature slightly higher than that maintained by the system of window ventilation. Our studies indicate that in order to obtain satisfactory results with this method of ventilation, the following conditions are essential:

(*a*) Fan and motor capacity, duct areas, and register areas should be ample—in accordance with the usual standards accepted in the best modern ventilating practice.

(*b*) It is highly desirable that the plenum air should be brought to each room by an individual duct originating at the heating chambers, the latter to

be constructed with hot air and tempered air chambers, with a connection from the individual duct to each chamber. In other words, the connections from the hot-air and tempered-air chambers should join immediately outside the heating chamber, with mixing dampers installed at the point of junction, these dampers being thermostically controlled by means of an intermediate acting thermostat serving this particular duct. A volume damper should also be placed in the duct for the regulation of the air quantities. This makes possible the regulation of air volumes and air temperatures according to the needs of each individual room, and also permits the opening of the windows in different rooms without seriously unbalancing the system as a whole.

(*c*) The incoming plenum air should be maintained at a moderately low temperature by means of the thermostats above referred to and the schoolroom should also be provided with direct radiation under both thermostatic and hand control sufficient only to counterbalance heat losses through walls, windows, floor and ceiling.

(*d*) In order to avoid objectionable drafts, within the zone of occupancy, it is desirable that the plenum air should be introduced to the room approximately 8 ft. above the floor, on an end wall of the room near the outside or window wall, if possible, with the exhaust outlet near the floor at the opposite end of the same wall.

(*e*) The exhaust ducts should be of ample size and should be provided with volume dampers and shut-off dampers. Aspirating cowls or ventilators are also desirable.

(*f*) A large thermometer clearly marked to indicate 70 deg. fahr. (21.1 deg. cent.) as a danger point should be conspicuously displayed on the teacher's desk.

Our studies of ventilation by the use of plenum and exhaust fans in combination have not been sufficiently extensive to justify conclusions as to the desirability of installing exhaust as well as plenum fans. The exhaust fans have the advantage of maintaining a constant exhaust co-related to the air supply. The recommendations applied above to plenum supply and gravity exhaust systems would also apply to the plenum supply and fan exhaust system.

CONCLUSIONS IN REGARD TO THE PRACTICAL RESULTS OF THE HUMIDIFICATION OF SCHOOLROOM AIR

In view of the prevalent conception that the dry air of the plenum ventilated schoolroom exerts a seriously harmful influence on health—a view supported by the interesting statistical studies of Ellsworth Huntington on the relation of climate and season to disease—we have devoted considerable attention to the observation of the practical effects produced by the washing and humidification of plenum air Two groups of children were exposed during the school hours of a period of sixteen weeks respectively to plenum air with an average relative humidity of 28.7 per cent, and to humidified air with an average relative humidity of 42.3 per cent. Careful observations of the physical development of the pupils at the beginning and end of the experimental period and detailed examinations for the detection of physical defects revealed no difference between the two groups. Exhaustive psychological studies of mental attainments and progress in school work again failed to reveal any difference between the two groups. From the standpoint of the incidence of respiratory disease during the period of observation the children exposed to humidified air showed an excess of respiratory disease among pupils in attendance but materially less absence due to respiratory disease as compared with the children exposed to ordinary plenum air. It is doubtful whether these differences are statistically significant.

While these observations of physical and mental conditions of pupils indicated no material influence of humidification, our experience indicated that the air of the humidified room was much less agreeable, as indicated by the opinions of teachers and observers than the air of the unhumidified room, being characterized by a more frequent occurrence of odor and stuffiness.

The lack of harmony between our results and the statistical studies of Professor Huntington may perhaps be due to the fact that the experience of eighty-six children for sixteen weeks gives too little material to reveal any objectionable effects which may result. On the other hand, however, it may well be that the excess death rate observed by Professor Huntington in dry winter weather is related to some factor which is not at all affected by the maintenance of a practical degree of humidification during the schoolroom hours. In any case the data at our disposal suggests that artificial humidification does not produce any striking improvement in the health or efficiency of school children.

VENTILATION OF THE SCHOOLROOM BY THE USE OF RE-CIRCULATED AIR

Modern views in regard to the supreme importance of temperature control in ventilation have on the one hand favored a tendency toward window ventilation with its diminished rate of aeration, and on the other hand have suggested the possibility of adapting the plenum system to the practice of re-circulation, the air exhausted from the schoolroom being washed and adjusted in temperature and humidity before being returned to the rooms.

Our studies of this method at the Y. M. C. A. College Gymnasium in Springfield, Mass., and in School 51, New York, have indicated that the practice of re-circulation is a highly economical one, making possible the saving of about half the fuel ordinarily consumed for plenum ventilation during the five colder months of the year. Furthermore our studies at School 51, during two successive seasons and under very careful control conditions, indicate with some clearness that the use of re-circulated air exerts no harmful effects upon the health or mental progress of the pupils in schoolrooms ventilated by this method.

On the other hand our experience with this ventilation procedure has been highly unsatisfactory from the esthetic standpoint. Re-circulation requires air washing with the maintenance of a relatively high humidity, a condition which in the previous experiments directed particularly toward the question of humidification we found to produce somewhat unsatisfactory results. As compared with plenum ventilated rooms at the same temperature and humidity, the general sensations of the occupants in regard to comfort were distinctly unfavorable to re-circulation, both in Minneapolis and in New York, and it was exceedingly difficult to persuade the teachers to endure the practice.

In the course of our experiments our own engineer at the school was constantly occupied in the regulation of the apparatus and in the attempt to keep conditions comfortable. Yet the air of the experimental rooms, and particularly the room supplied with re-circulated air, seemed so stale and unattractive that only the interest of the teachers in the ultimate objects of the investigation reconciled them to its continuance. It was hardly possible for any unprejudiced observer to pass from an ordinary fresh-air window-ventilated room to either of the plenum ventilated and humid experimental rooms without noting the distinctly unfavorable impression which was received. If such was the case under a system of careful and continuous expert supervision, it seems to us probable that the practice of re-circulation as operated by the ordinary janitor would be likely to lead to very great dissatisfaction.

The re-circulating system with air washer (in which provision is made so that a mixture of the re-circulated and fresh outside air is possible), may be found to be of value under certain conditions. This would be the case in buildings of loose construction or great leakage of air, in rooms of limited occupancy, or in buildings in which the total occupancy is relatively small in proportion to the total cubic contents of the building. Re-circulation may also well be used for pre-heating, i. e., prior to the period of occupancy. In the dwelling house the connection of the fresh-air intake of the furnace with the front hall of the house is a most excellent device for the saving of fuel. In the schoolroom, however, while re-circulation does not seem likely to damage the health of the pupils, the possibilities of discomfort and complaints deter us from recommending this method as a general procedure.

SPECIFIC CHARACTERISTICS OF WINDOW VENTILATION AND FAN VENTILATION AS REVEALED BY THE PRESENT STUDY

Our investigations have suggested that the two alternative procedures which may be considered generally satisfactory for the ventilation of the schoolroom are (*a*) ventilation by window inlets with gravity exhaust, and (*b*) ventilation by plenum fans. It remains to consider in detail the characteristic differences in the air conditions maintained by these two procedures and their effects upon the health and comfort of the pupils.

The first difference between these two methods of ventilation lies in the less complete aeration attained by window ventilation. Our results are uniform in indicating that the carbon dioxide values will be somewhat higher with window ventilation and this result is an expression of the fact that the air change provided by this method falls materially below the 30 cu. ft. (0.85 cu. m.) per min. which in the past has been accepted as essential. Furthermore in fan ventilation the air passes across the top of the room and back along the floor in the opposite direction,—while in window ventilation the direct air flow is across the top of the room and from the center to the inner wall, with no strong current in the zone of occupancy. In view of the modern views of ventilation as primarily a procedure for maintaining desirable thermal conditions, the diminished aeration seems to be unobjectionable, and our results indicate that aeration diminished in this way to a reasonable degree and accompanied by little or no air flow near the floor is not accompanied by the production of offensive odors and is beneficial rather than harmful to the health and comfort of the occupants.

The second difference, correlated closely with the first, is to be found in the fact that window-ventilated rooms are characteristically maintained at a temperature slightly lower than fan-ventilated rooms. Considering the large number of schoolrooms considered in the course of our investigation, it seems reasonably clear that, in a window-ventilated room with a comparatively slight air movement, temperatures between 66 deg. to 68 deg. fahr. (18.9 deg. to 20 deg. cent.) are borne with comfort; while in the fan-ventilated room with the increased heat loss due to greater air movement the temperature must be maintained between 68 deg. and 70 deg. fahr. (20 deg. and 21 deg. cent.). Even under such conditions the rate of heat loss as measured by the kata-thermometer is greater in the fan-ventilated room, but our results indicate that the kata-thermometer somewhat exaggerates the effect of air movement upon the clothed body.

Finally it appears from our studies with specially designed air thermometers that a temperature in the window-ventilated room is characterized by continuous slight fluctuations as contrasted with a relative constancy of temperature maintained in the fan-ventilated room.

The choice between these two methods must in general, therefore, rest (1) upon the relative desirability of surrounding the body with a rapidly moving current of warmer air or a more moderate current of cooler air, and (2) upon the relative course of construction and operation of systems of the two types.

THE SENSATION OF COMFORT EXPERIENCED RESPECTIVELY IN WINDOW-VENTILATED AND FAN-VENTILATED ROOMS

As indicated above, the choice between window and fan ventilation as general methods for schoolroom ventilation must be based largely upon comfort of the occupants and the influences of the two methods upon respiratory disease, since we have been unable to detect any significant influences exerted by ventilation procedures upon general physical health or mental progress.

In regard to the first of these standpoints, that of the comfort of the occupants of the schoolroom, we have obtained a considerable volume of data under widely varying conditions. The comfort votes of the teachers and of our own observers may in given instances be influenced by prejudice or by intrinsic local variables, but on the whole the recorded differences are so striking as to furnish fairly

conclusive evidence of a tendency to prefer the cooler window-ventilated room to the warmer fan-ventilated room.

In the investigation of 1916-1917, which is the most intensive and prolonged study we were able to conduct, it was made clear neither fan nor window ventilation produces satisfactory results when both types of rooms are overheated (School 33); that when fan rooms are maintained at a lower temperature than window rooms (School 97) the fan rooms are distinctly preferred; but that when, as is normally the case, the fan-ventilated room (School 115) averages half a degree to a degree higher than the window-ventilated room there is a distinct preference expressed for the latter.

It seemed probable to us from these studies that the strong, steady currents of air at a uniform temperature, characteristic of fan ventilation, may be actually less desirable than a less intense but more fluctuating air flow of slightly varying temperature. The strong currents produced by the fans tend either to produce drafts on the exposed parts of the bodv, or to lead to the demand for an increase in room temperature which makes the face comfortable but deprives the membranes of the nose and throat, and perhaps also the clothed parts of the body, of a desirable stimulus of coolness. It seems probable that a much less active general air movement, accompanied by minor slight fluctuations (such as we find to be characteristic of the window-ventilated room) is actually more agreeable and more stimulating to the vaso-motor system.

In order to test this matter more fully, we conducted an exhaustive series of tests at the Wadleigh High School, in which votes were obtained from a group of high school girls who were blindfolded and led from room to room, so that it was impossible for them to have any idea as to the method of ventilation maintained, and who were therefore entirely uninfluenced by any preconceptions in regard to the matter. As usual the window rooms in this study averaged a trifle cooler than the fan rooms and showed a greater variability; relative humidity was practically the same in the two groups; the deheating effect as usual was greater in the fan room, and aeration materially greater. The votes of the subjects indicate that the window rooms appear fresher and have less odor, that they are much less stuffy, they are less drafty and yet more breezy (that is, that they possess appreciable air movement that is not annoying), and that they are generally more comfortable.

There are several differences in the mode of heat loss from the body in the two types of rooms which may help to explain the sensation votes recorded. In the first place the window rooms show greater differences between the temperature at the floor and at the 3-ft. level. It is conceivable though not probable that such a condition is agreeable. In the second place heat loss in the window rooms by radiation and conduction is relatively greater, heat loss by convection relatively less. The nose and throat are thus bathed in cooler air and the rate of heat loss from the clothed body as compared with the unprotected face must be relatively greater. This may very probably be a condition favoring comfort. In the third place the fact that the air movement in the case of the window rooms is more variable from moment to moment may be a factor which plays a part by producing the pleasant sensation of "breeze" without causing "drafts."

That the window-ventilation system shows a real superiority in these respects, however, appears quite certain.

THE PREVALENCE OF RESPIRATORY DISEASE IN WINDOW-VENTILATED AND FAN-VENTILATED ROOMS

Finally in the study of the prevalence of respiratory disease in window-ventilated and fan-ventilated rooms, conducted in co-operation with Dr. S. Josephine Baker of the New York City Department of Health, we have obtained our most striking and important evidence in regard to the relative merits of the two procedures. Careful observations have been made of the absence due to respiratory illness and of the respiratory illness among pupils in attendance in groups of representative schoolrooms of three types: (a) window-ventilated rooms with

gravity exhaust maintained at temperatures averaging about 59 deg. fahr. (15 deg. cent.); (b) window-ventilated rooms with gravity exhaust maintained at temperatures averaging 66 deg. to 67 deg. fahr. (18.9 deg. to 19.4 deg. cent.); and (c) plenum ventilated rooms with gravity exhaust maintained at temperatures between 68 deg. and 69 deg. fahr. (20 deg. to 20.6 deg. cent.). The study covered eight weeks in the late winter of 1916 and twelve weeks in the autumn and winter of 1916-1917, 2,500 children being included in the first study and 3,000 in the second.

Both of these studies indicated approximately the same amount of respiratory illness in the two types of window-ventilated rooms, irrespective of a difference of temperature of 7 deg. or 8 deg. fahr. (3.9 or 4.4 deg. cent.), and the concordance of these results is strong evidence of the reliability of the methods used in this study. On the other hand the warmer fan-ventilated rooms showed on the average 18 per cent more absences due to respiratory illness and 70 per cent more respiratory illness among children in attendance, in spite of the fact that per capita floor space was greater in the fan-ventilated rooms, than in the window-ventilated rooms.

These results in our judgment strongly suggest that the slightly higher temperature and more uniform air conditions characteristic of fan-ventilated rooms exert a real and important influence in promoting susceptibility to respiratory disease.

GENERAL CONCLUSIONS IN REGARD TO THE IDEAL METHOD OF SCHOOLROOM VENTILATION

In final summary we may conclude that either window ventilation or plenum fan ventilation,—if the plant be properly designed and operated,—yields generally satisfactory results from the standpoint of the air conditions in the average schoolroom. We have found it possible to maintain by either of these procedures air conditions in the schoolroom that would be considered satisfactory by all the ordinary physical tests and conditions which are reasonably comfortable and satisfactory to the occupants. The main difference lies in the fact that the air of the fan-ventilated room is likely to be about 2 deg. fahr. (1.1 deg. cent.) higher, and more uniform in temperature, and that the air movement in this method is considerably greater.

We find that on the whole, other things being equal, the window-ventilated room at 67 deg. fahr. (19.9. cent.) is somewhat more comfortable than the fan-ventilated room at 69 deg. fahr. (20.6 deg. cent.).

We find that the window-ventilated room at 67 deg. fahr. (19.4 deg. cent.) is characterized by materially lower incidents of respiratory disease than the fan-ventilated room at 69 deg. fahr. (20.6 deg. cent.).

It appears that 68 deg., fahr. (20 deg. cent.) is a critical temperature as regards both comfort and susceptibility to respiratory disease; and the great advantage of window ventilation lies in the fact that the reduced air flow in the zone of occupancy permits this low temperature to be maintained.

So far as cost is concerned we have made no detailed investigation. It seems reasonably certain, however, that the cost of extra radiation and window boards in the window-ventilated room would be more than balanced by the ducts and fans required in the plenum method, and from the standpoint of operation the lesser aeration in the window-ventilated room must certainly imply a substantial diminution in coal consumption. During 1914-1915 the Sherman School at Fairfield burned less than 7 tons of coal per classroom against over 10 tons per classroom for the Bridgeport fan-ventilated schools.

In general, therefore, we are forced to conclude that window ventilation with ample direct radiation, window deflectors and adequate gravity exhaust, seems the most generally promising method for the ventilation of the classrooms where local conditions permit its use. We have not made extensive studies of the results to be attained by supplementing either window or plenum fan ventilation by fan exhausts. Such data as we have obtained are not particularly promising along this line. The object of the exhaust fan is to promote a more rapid and more constant air flow, and the general tendency of our results is to suggest that it is the very desire to promote a large and constant air flow that leads to the deficiencies

displayed by plenum ventilation. If our general conclusions are justified, the ideal which has been set before the ventilating engineers of a constant air flow of 30 cu. ft. (0.85 cu. m.) per min. is an excessive one, and the aim should be to maintain not a large flow of warm air but a relatively less rapid flow of air at a lower temperature.

It may well be that the art of plenum ventilation can be developed along the newer lines indicated by these conclusions so as to furnish interesting and valuable results. Experiments along this line should contemplate a reduction in the amount of air flow now provided by ordinary ventilating standards, with the introduction of air at a materially lower temperature. If this were done with the ordinary location of inlets and outlets the cold air would tend to drop to the floor and produce unpleasant drafts. We are inclined to believe, however, that an essential value of the window-ventilation method lies in the fact that a slightly variable mixture of warm and cool air rises from radiators and windows and mixes gently with the general air of the room.

THE IMPORTANCE OF CONSIDERING LOCAL CONDITIONS IN THE DESIGN OF VENTILATION EQUIPMENT

The most harmful influence in the history of the art of ventilation has been the tendency to adopt certain standardized procedures and apply them without special study to individual cases. It would be most unfortunate if the results obtained by the Commission should be interpreted as a plea for the uniform routine application of the system of window ventilation described above. We believe that this method shows great promise and will perhaps be found more frequently useful than any other procedure for the air conditioning of the average classroom. On the other hand there will be many schools in which this method can not be applied. Where noise, and dirt, and odors from the streets make it inconvenient to have windows open the use of the plenum system would, for instance, be clearly indicated; and where classrooms are markedly overcrowded, window ventilation would be likely to prove inadequate. The arrangement of plan and the construction of some large school buildings would make impossible the application of this system.

Above all it should be emphasized most strongly that our conclusions apply only to the schoolroom and that other types of buildings would require different treatment. The hospital ward can apparently be ventilated satisfactorily by the use of windows alone without the use of exhaust ducts. The schoolroom requires the installation of exhaust ducts in order that desirable air conditions may be maintained, but the fresh air supply may, as our studies indicate, be obtained directly from the windows. In crowded auditoria and factories, on the other hand, this method would generally prove unsatisfactory, and for such occupied spaces fan ventilation is likely to supply the only adequate solution of the problem.

AVOIDANCE OF OVERHEATING THE ONE UNIVERSAL ESSENTIAL

Finally it must be emphasized in closing this discussion that the avoidance of overheating is the primary essential in all systems of ventilation. Air change, direction of flow and all other factors are secondary. The most important article of ventilating equipment is the thermometer; and however simple or however complex an apparatus may be installed for air conditioning, a constant and intelligent vigilance in regard to operation and overheating is the price of health and comfort.

DISCUSSION

WHEN this matter was introduced at the Semi-Annual Meeting 1923 of the Society in Chicago, President H. P. Grant pointed out that the Publication Committee wished to bring the general conclusions of the report to the attention of the membership and stated further that due to the fact that we are a Society of technical men, "we are in no way interested in the practical devices or

methods used to accomplish the given results in ventilation of schools but we are interested in the results obtained."

S. R. LEWIS: I should like to present some of my experiences on this subject.

As I understand it, the New York State Commission on Ventilation has advocated an arrangement of hooded inlets, preferably at the windows, with direct radiators under the windows, and with outlets from the rooms to flues which are arranged for gravity or mechanical eduction.

I studied such a plant as this in sub-zero temperature in northern Michigan. The rooms had thermostats to control the direct radiators; the inlets were effectively hooded and screened, in the lower sashes of the windows, and there were electric fans to the attic which insured a positive suction in the outlets from the rooms.

For a few minutes after school opened the sense-impression was excellent. One felt the fresh, cool air in his face, the floor was warm, and the distribution of the entering air was favorable. The heat from the bodies of the occupants and the bright sunshine, however, soon caused the thermostats to act on the steam valves, shutting off the radiators. Immediately the icy entering air, no longer buoyed up by the rising currents of hot air from the radiators, dropped to the floor and was felt on one's feet. The teacher in self-defense soon closed the inlets.

The amount of direct radiation was necessarily in excess of that necessary to maintain the temperature, with inlets closed, having been calculated to heat the entering cold air. Thus, if the thermostat could be prevailed upon to open the steam valves, they were open only for a very short time, and the teacher finally opened the corridor door. The open corridor door served as an inlet to supply the air exhausted by the fan, drawing air via the corridor from locker rooms, toilets, etc. This air short circuited from the door to the vent flue, leaving the greater part of the room unventilated, over-heated and unsatisfactory for the balance of the day. When the corridor doors were open the warm air from the corridors tended to keep the themostats warm, keeping the radiators in the rooms cold.

The only recourse, apparently, was to raise some windows, chill the the thermostat and room, so as again to heat the radiators, when for a time the corridor door could be closed, the window inlets could be opened, and the cycle could be repeated.

There was complaint that pipes were frozen frequently, due to the failure of the janitor each night to visit and close by hand the hundreds of little steel shutters in the inlets, and due to corrosion of these inlets, as well as to obstruction of the shutters by dirt or ice.

My conclusions from studying this building in operation are:

1. Exhaust ventilation with gravity inlets is impractical because of difficulty in keeping the designed inlets open, and because of difficulty in keeping doors which cause short circuits closed.

2. Introduction of cold air direct from outside is unsatisfactory because toleration of such introduction depends on having local heaters heated while such introduction of cold air proceeds.

No occupied classroom can possibly have its direct radiators heated continuously without excessive temperature, with attendant discomfort and waste.

3. It is not economical or wise to depend on control by a very busy teacher of numerous air inlets and radiators. Varying conditions of wind, sunshine and occupancy render the manual control of direct radiation with window inlets and exhaust ventilation an arduous task, leaving the teacher little time to devote to educational duties.

If the plant described is an example of the type of schoolroom ventilation recommended by the New York State Commission on Ventilation, I submit that no remarkable or epoch-making improvements have been discovered.

E. S. HALLETT: The report of the New York State Commission on Ventilation gives the impression that the commission composed of a number of physicians and one engineer have discovered more heating and ventilating data in three years than all of the technical and practical engineers of the world have done in all time, when as a matter of fact there is much experimental and practical data on record, which is of more recent origin than the broad and general conclusions stated in this report.

The report criticizes the use of too high temperatures in rooms ventilated by mechanical means, but this only exists when the heat control is not functioning or where there is insufficient air motion.

It has recently been scientifically proven at the Research Laboratory in Pittsburgh, that the temperatures for comfort and health are variable and depend upon the humidity and air motion. There is abundant evidence that exhalations of the breath and body are injurious to occupants of a room, and it is only possible to overcome this condition by a complete mechanical system which includes an air washer and ozone apparatus. If the tests using open ventilation alone had been conducted on average winter days not a teacher would have failed to close the window or vacate the room.

The Commission found that the most popular ventilation system is window inlets with gravity exhaust. With this plan circulation would depend upon wind velocity and temperature and the air motion would be confined to a belt across the room opposite the window. Furthermore, children next to the radiators would be uncomfortable, carbon dioxide readings (CO_2) would fluctuate greatly, humidity would be low and uncontrolled. The exhaust in this case would have to be 4 times as large as needed with the plenum system.

In enumerating the conditions essential to the success of the window inlet and gravity exhaust system, the commission points out that "a large thermometer with 68 deg. F. clearly indicated as the danger point should be displayed in a prominent position on the teacher's desk." A heating and ventilating system worthy of the name should not be a distraction and interfere with both teachers' and pupils' work. Under present heat-

ing and ventilating standards nothing indicating the presence of the heating system appears in the room, which is as it should be.

The criticism of the plenum gravity exhaust system is of little significance in the light of practical experience, especially as regards hand control of radiators. On the item of humidification the Commission declines to bring in any findings in view of Dr. Huntington's studies and it also ignores the more recent studies of the relation of humidity to health and comfort made by the Society's Research Laboratory.

The Commission's studies of re-circulation seem to be prejudiced though St. Louis schools are a good example of the fact that as much as 90 per cent of the air can be re-circulated without any comment or ill effects. Furthermore, it has been found that air discharged from vent stacks of St. Louis schools was cleaner than air entering intakes.

The Commission's findings, that open window tests give 30 per cent better showings on respiratory diseases, is not borne out by scientific tests or records, in fact, Detroit's experience has been to the contrary. The fact that defective equipment was used in these tests makes them misleading.

The writer of the Summary for the Commission admits that where noise, dirt and odors from the streets make it inconvenient to open windows, the plenum system would be preferable. Also in cases of overcrowding which is an admission that where a real case of ventilation is required the plenum system must be used. It must follow also that if the plenum system is necessary in difficult cases, it must also be superior in all cases.

C. L. RILEY: This Commission, while it was a State commission in the sense that it was appointed by the Governor of the State of New York, did not operate on State funds, but on private capital that was furnished for the purpose. When this capital gave out, the work of the Commission ceased. The reason the results are so peculiar, I think, is because the work was never finished.

If you read Section 2 on Schoolhouse Ventilation, you will find that while they laid out ten different schemes of gravity, window and fan ventilation, getting every possible combination, they finally narrowed down in their work, to practically two schools with two systems; the window inlet and gravity outlet, and the plenum inlet and gravity outlet. They did not, therefore, cover the ground thoroughly. The statement of their results, even in their own words, is doubtful and tentative. They were not sure what they were talking about and they knew it.

In the entire 620 pages of the report we do not find any consideration of the practical details of operating a ventilating system, and the conclusions are so tentative and so lacking in conviction that I cannot avoid a feeling of surprise that the Commission should have rendered a report of this incomplete character. For instance, under the heading, "Conclusions," in regard to the practical results of the humidification of schoolroom air, the concluding paragraph reads as follows: "The lack of harmony between our results and the statistical studies of Professor Huntington may perhaps be due to the fact that experience of 86 chil-

dren for 16 weeks gives too little material to reveal any objectionable effects which may result. On the other hand, however, it may well be that the excess death rate observed by Professor Huntington, in dry winter weather is related to some factor which is not all affected by the maintenance of a practical degree of humidification during the schoolroom hours. In any case the data at our disposal suggest that artificial humidification does not produce any striking improvement in the health or efficiency of school children."

Now what kind of a report is this, that humidity does not produce any marked effect? If their experiments had been of any conclusive character, they could not have reported in such indefinite terms about them. They would have shown an effect one way or the other, and from that we can take it that they themselves admit that their report is inconclusive.

This matter is of considerable importance to the profession because it is receiving wide publicity, and we should handle it with some care. It occurs to me that we should find some way of suggesting to this Commission that, inasmuch as they had to leave their work in an unfinished condition, this Society will be glad to go on and assist them in finding conclusions based upon what they have done and what may be done in the future, which will accord with a real thorough analysis of the subject. This seems to me one way of turning this report into something constructive.

J. I. LYLE: In reference to Mr. Hallett's proposal that a Committee be appointed to report before our next Annual Meeting, I think it would be a great mistake. The mistake of the New York Ventilation Commission has been in turning out a half-baked report. Their mistake has been in taking a lot of data with a lot of opinions of which there is no scientific basis presented. There are no readings or facts to support their opinion. It would be the greatest mistake for this Society to follow in their footsteps and have a committee rush through a report. Then, again, we cannot have a committee publish a report that is not passed upon by our body as a whole. It will have to come before either our Annual or Semi-Annual Meeting and we don't have another meeting until next January.

It isn't a question of days or weeks or months. It is a question of years. These things don't force an upheaval in one year. A lot of people will rush into this thing and a lot of them will rush away from it.

Now let's not rush at it. Let's appoint a proper committee, a studious committee, to confer with the New York State Commission, give them every opportunity to justify the conclusions which they have given in this book, and then make a sane report, constructive as we can make it, making use of everything we can that is in this report that is of use and discard the rest. The school buildings are not all going to be built in the next twelve months, and of those that are built they are not all going to have window ventilation. I think it will have very little effect on results, but the worst mistake we could make would be in taking hasty action.

DR. E. VERNON HILL: I don't think the report is worth the appointment of such a committee, as this would be duplicating the work the

Society has been doing for years, it will be duplicating and checking upon the work of the Research Bureau, and it will really amount to questioning our own position and the objects and ideals which we have. The work which this Commission has done is work which we did years and years ago.

I have been thrown into contact time and time again with men who become interested in ventilation, particularly school ventilation. Here is about the way it works. First, he becomes convinced that the only thing is open ventilation, "God's fresh air,"—but after a while he learns that there is some defect in the window ventilation scheme. After an unsatisfactory trial he tries to improve it by adding a vent flue, but finds that it doesn't always operate as he thought it would. Next, he sticks an aspirating coil in the vent flue. That is better, but the aspirating coil isn't just what he wants so he installs a fan. The result is window ventilation with a fan exhaust. As he continues, he finds defects in this plan because the exhaust system is not under control. When he gets the exhaust under control he finds that the air supply must be regulated. Going a step farther he finds the necessity for temperature control, humidification, air cleaning, and as a result he discovers that he has a fairly good ventilating system. We know about all this—we have done all of it—it's history now. Our next step is a correct conception of air conditions, the relation of temperature, and humidity.

Now if the Commission had stuck to its job, it would have gone along and finally have stood probably where we now stand. If we have a big Committee to go back all over this problem, we will have to educate the New York State Commission, and after years of time we will bring it up to the point where we are at the present time.

Homer Addams presented a motion that the report and its discussion be referred to a suitable committee of the Society for consideration conference and recommendations to be presented at the next Annual Meeting, and after considerable discussion on this motion by Messrs. Hallett, Allen, Lyle, Hart, Linn, Howatt, E. Vernon Hill, Riley, Chapman and Lewis, the motion was passed.

No. 661

INSULATION OF COLD SURFACES TO PREVENT SWEATING

By Leonard L. Barrett[1], New York, N. Y.

Member

IT is the common practice to insulate cold surfaces with hair felt, cork, or similar substances for the purpose of preventing sweating even when the question of the heat gained by such surfaces is of no economic importance. Such insulations are frequently applied without proper consideration of the conditions which will surround their use and in such cases they often fail to give satisfactory service. Sweating may occur on the surface of the insulation due to insufficient thickness or it may occur on the cold surface itself due to the fact that the insulation is not air-tight. It is the object of this paper to establish a mathematical basis for determining the thickness of insulation required to prevent sweating on the surface of the insulation under various conditions of temperature and humidity, and to present curves which will obviate the necessity of any mathematical work in making practical determinations.

Moisture will be deposited on a surface whenever its temperature falls to that of the dew point. Fig. 1 gives the temperature difference between the air and the dew point corresponding to different air temperatures and different relative humidities, as computed from Goodenough's tables.

MATHEMATICAL ANALYSIS—FLAT SURFACES

Let T° = temperature of the air on warm side of insulation

T' = temperature of the dew point of surface temperature on warm side of the insulation when sweating begins

T'' = surface temperature on cold side of insulation

T''' = temperature of the medium on cold side of insulation

H = heat gained per sq. ft. per hr.

k = conductivity of insulation, B.t.u. per sq. ft. per hr. per in. thick per deg. temperature difference between inner and outer surfaces of insulation

[1] Manager, Engineering Department, Keasbey & Mattison Co.

Paper presented at the Semi-Annual Meeting of the American Society of Heating and Ventilating Engineers, Chicago, Ill., May, 1923.

t = thickness of insulation required to prevent sweating, in.

c = surface transmission coefficient on warm side of insulation B.t.u. per sq. ft. per hr. per deg. fahr. temperature difference between surface of insulation and air.

c' = surface transmission coefficient on cold side of insulation, in same units as c.

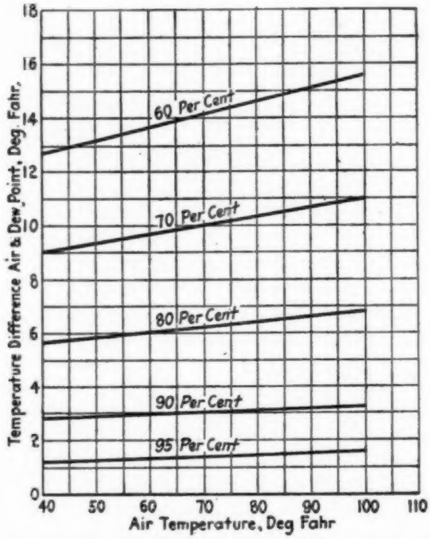


FIG. 1. TEMPERATURE DIFFERENCE BETWEEN AIR AND DEW POINT AT DIFFERENT RELATIVE HUMIDITIES PLOTTED AS A FUNCTION OF AIR TEMPERATURE

The following equations can be written at once:

$$H = c\ (T^\circ - T') \tag{1}$$

$$H = \frac{k\ (T' - T'')}{t} \tag{2}$$

$$H = c'\ (T'' - T''') \tag{3}$$

Eliminating T'' from (2) and (3) gives $H = \dfrac{T' - T'''}{\dfrac{t}{k} + \dfrac{1}{c'}}$ (4)

Equating (1) and (4), solving for $T' - T'''$, adding $T^\circ - T'$ to each side of the resulting equation, and solving for t, it is found that

$$t = \frac{k}{c}\left[\frac{T^\circ - T'''}{T^\circ - T'} - 1\right] - \frac{k}{c'} \tag{5}$$

When the value of c' is very great, as is the case when there is a very high air velocity on the cold side of the insulation, or when the cold surface of the insulation is separated only by a metal wall from a cold liquid such as cold water or brine, T''' and T'' become practically equal, and (5) becomes

$$t = \frac{k}{c}\left[\frac{T^\circ - T'}{T^\circ - T'''} - 1\right] \tag{6}$$

PIPES

In the case of pipes containing cold water, brine, or other cold liquids, T'' is practically equal to T''' and the value of c' is so great as to make it unnecessary to complicate the deduction of the equations by the introduction of an equation giving the heat gained in terms of c' and $T''-T'''$.

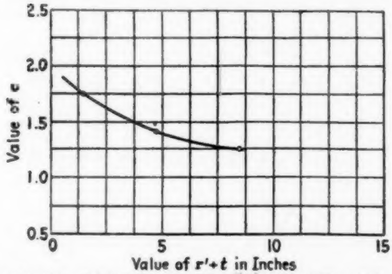


FIG. 2. TRANSMISSION COEFFICIENT FROM AIR TO COVERING AS A FUNCTION OF OUTER RADIUS OF COVERING

Assume a pipe of outside radius r' (in.) encased in an insulating covering of thickness t. Let H' be the heat gained per linear foot of pipe per hour, and let the other symbols have the same meanings as heretofore. Consider a cylindrical shell of the covering having a radius of r (in.), a thickness of dr and a difference of temperature between its inner and outer surfaces of dT. Now since the heat passing through the insulation is directly proportional to the area through which it flows and inversely proportional to the length of the path, we have

$$H' = \frac{2\pi r\ k}{12}\ \frac{dT}{dr} \tag{7}$$

Integrating between temperature limits of T' and T''' and covering radii of r' and $r' + t$, we have

$$H' = \frac{2\,\pi k\,(T' - T''')}{12 \log_e \dfrac{r' + t}{r'}} \tag{8}$$

Consider now the heat gained as being gained by transmission from the air to the outer surface of the insulation, giving

$$H' = \frac{2\pi c}{12}(r' + t) \quad (T^\circ - T') \tag{9}$$

Equating (8) and (9), solving for $T' - T'''$, and adding $T^\circ - T'$ to each side of the resulting equation, the result is

$$T^\circ - T''' = (T^\circ - T')\left[\frac{c}{k}(r' + t)\log_e \frac{r' + t}{r'} + 1\right] \tag{10}$$

This equation gives the temperature difference between pipe and air at which sweating will occur when the thickness of the covering and the temperature difference between the air and the dew point are known or assumed.

It is interesting to observe that equations (6) and (10) are identical when $r' = \infty$, which is the case for a flat surface; the value of the shape factor $(r' + t)\log \frac{(r' + t)}{r'}$ in equation (10) becoming equal to t in this case. This is not immediately obvious as the shape factor assumes an indeterminate form when $r' = \infty$ is substituted therein. However, upon differentiating the shape factor with respect to r', placing the derivative equal to zero, multiplying the resulting equation through by $r' + t$, and then placing $r' = \infty$, the shape factor becomes determinate and equal to t.

VALUES OF c AND k

Heilman[1] has experimentally established the curves of heat loss per square foot of canvas covered surface of insulation as a function of $T^\circ - T'$ in the case of pipes having diameters of the outer surface of the covering of 3.1 in., 9.5 in., and 17.2 in., respectively. The values of c given by these curves for a temperature difference of 10 deg., and which will not vary materially for temperature differences $(T^\circ - T')$ from 0 to 20 deg. are: 1.74 for 3.1 in. O.D., 1.4 for 9.5 in. O.D., 1.25 for 17.2 in. O.D. These values have been used in Fig. 2 to establish a curve giving values for c as a function of $r' + t$.

The values of k corresponding to the various coverings in common use, as determined by Van Dusen[2], are given herewith:

TABLE 1. VALUES OF K CORRESPONDING TO COMMONLY USED COVERINGS

	k	Per Cent Variation from k=0.3
Hair Felt	0.246	—18%
Keystone Hair	0.27	—10%
Wood Felt	0.36	+20%
Cork, 6.9 lb. per cu. ft.	0.27	—10%
Cork, 9.7 lb. per cu. ft.	0.30	0%
Cork, 16 lb. per cu. ft.	0.35	+17%

[1] *Trans. A. S. M. E.*, 1922, Fig. 4.
[2] JOURNAL AM. SOC. HEAT.-VENT. ENGINEERS, Vol. 26, p. 646.

A value of 0.3 for k has been used in the computation of the curves given in this paper. As the thickness of covering required varies directly with the value of k for the covering used in the case of flat surfaces, and approximately so in the case of pipes, the thickness determined from the curves should be increased or decreased, according to whether the value of k is greater or less than 0.3, and by the percentage indicated in the last column of the foregoing tabulation.

DISCUSSION OF CURVES

The curves in Fig. 3 are plotted from equations (6) and (10), and

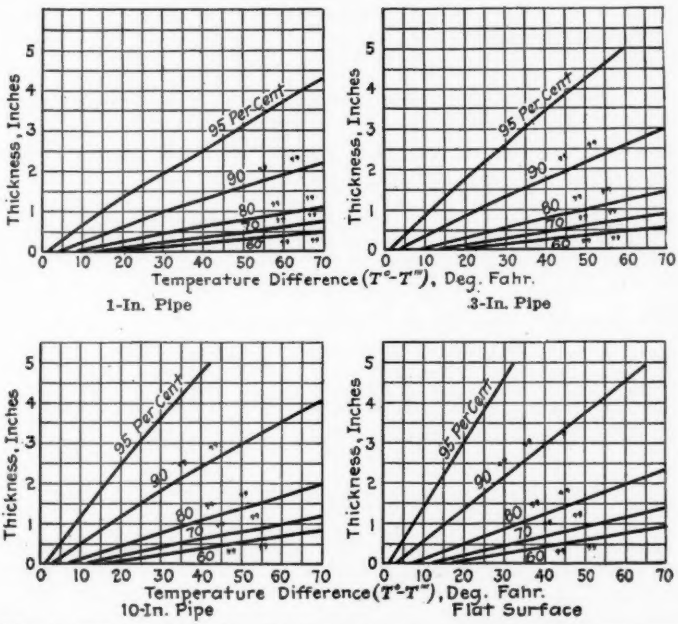


FIG. 3. THICKNESS OF INSULATION TO PREVENT SWEATING AT DIFFERENT RELATIVE HUMIDITIES PLOTTED AS A FUNCTION OF TEMPERATURE DIFFERENCE BETWEEN COLD MEDIUM AND STILL AIR

using the values of $T°—T'$ corresponding to the assumed humidities as given in Fig. 1 for an air temperature of 80 deg. fahr. It will be noted from Fig. 1 that the value of $T°—T'$ corresponding to any relative humidity varies but slightly with the air temperature for the air temperatures ordinarily encountered on the warm side of the insulation when sweating occurs. This makes it possible to disregard the air temperature in Fig. 3 and to label the curves with the humidities instead of with the corresponding values of $T°—T'$.

Curves are given for three different pipe sizes, for the greater the pipe size, the thickness of insulation and all other conditions remaining constant, the less will be the temperature of the surface of the covering, and the greater will be the insulation thickness necessary to keep the surface temperature above the dew point. The curves are straight lines in the case of the flat surface as will be evident from equation (6).

None of the curves pass through the origin because no insulation is required when the temperature difference between cold surface and the air is less than the temperature difference between the air and the dew point. For instance, at humidity 80 per cent the temperature difference between the air and dew point is 6.5 deg., therefore no insulation is required until the temperature difference between the cold surface and the air is equal to 6.5 deg.

EXAMPLE OF METHOD OF COMPUTATION OF CURVES

Let it be required to determine a point on the 95 per cent humidity curve for 3 in. pipe and assuming a covering 1 in. thick.

$$(r' + t) \log_e \frac{(r' + t)}{r'} = 2.75 \log_e 2.75/1.75 = 1.24$$

$k = 0.3$

$c = 1.6$ [from Fig. 2, (for $r' + t = 2.75$)]

$T^\circ - T' = 1.5$ deg. (from Fig. 1)

$$T^\circ - T''' = 1.5 \left[\left(\frac{1.6}{0.3} \right) (1.24) + 1 \right] = 11.4$$

EFFECT OF AIR MOVEMENT ON WARM SIDE OF INSULATION

The experiments of Heilman, from which the values of c are obtained were conducted under still air conditions, that is, where the air circulates due to natural convection currents only. If the air on the warm side of the insulation have sufficient velocity the values of c will be considerably greater than the values given in Fig. 2, and it is apparent from equation (6) that the greater the value of c the less will be the thickness of insulation required. Therefore the thicknesses given by the curves are maximum thicknesses and are applicable to still air conditions, as where the cold surfaces are located in confined spaces. It has seemed best to prepare the curves in Fig. 3 for still air conditions, as this condition gives the best standard of reference, and at the same time gives the worst conditions as regards the prevention of sweating. Furthermore, the values of c corresponding to this condition are well established by experiment.

If the cold surfaces are located where there is much air circulation on the warm side of the insulation, the thickness shown in Fig. 3 may be

reduced. The experiments of T. S. Taylor[4] have shown that the value of *c* increases approximately uniformly with the velocity at low velocities and that with an air movement of 800 ft. per min. the value of *c* will be 5.71. The value of *c* for still air found in the same determinations at a temperature difference of 10 deg. fahr. is 1.68. Since the thickness of insulation required for flat surfaces varies inversely as the value of *c*, it is a simple matter to utilize this data to establish a curve showing the percentage reduction for different air velocities to be applied to the values of *t* given in Fig. 3. This curve is given in Fig. 4. For practical purposes this curve can also be used in the case of pipes without any serious error.

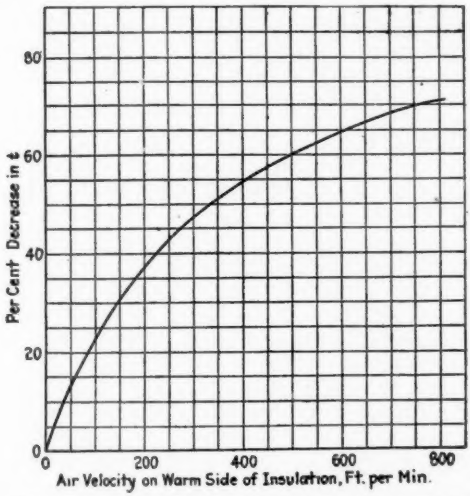


FIG 4. PERCENTAGE DEGREE AT DIFFERENT AIR VELOCITIES IN THICKNESS GIVEN BY FIG. 3 FOR FLAT SURFACES

It should be pointed out that corrections for the velocity of the air on the warm side of the insulation should be made with caution, and in most cases should not be made at all. Still air conditions will usually prevail at one time or another, and unless the insulation thickness required by such conditions is applied sweating will then occur.

INSULATION OF ROOFS AND EXTERIOR WALLS

When insulation is applied to roofs or exterior walls to prevent sweating the maximum effect of outside air velocity is to reduce the temperature of the cold side of the insulation T'' to the temperature of the out-

[4] JOURNAL AM. SOC. HEAT.-VENT. ENGINEERS, Vol. 26, p. 451. The factor for converting watts per sq. cm. per deg. cent. temp. dif. to B.t.u. per sq. ft. per deg. fahr. temp. dif. per hr. is 1764.

side air. Equation (6) is then applicable and the flat surface curves in Fig. 3 gives the proper thickness of insulation. For lesser outside air velocities the thickness of insulation could be somewhat reduced, but due to the fact that very high outside air velocities occur from time to time it is advisable where insulations are used on roofs or outside walls to prevent sweating to consider that T'' is equal to the temperature of the outside air. This will give a thickness suitable for the most extreme velocities. Correction for inside air motion would be made in such cases in the manner described in the preceding paragraph. The thickness required may also be reduced according to the insulating effect of the structural materials entering into the construction of the wall or roof. For instance, if the wall is faced with 8 in. of brickwork of conductivity $k = 5$, the thickness of insulation given by the curves may be reduced by $0.3 \times 8/5 = 0.48$ in.

INSULATION OF COLD SPACES

When it is desired to insulate cold spaces containing air at a definite low temperature, such as refrigeration rooms, in such a way that there shall be no condensation on the warm side of the insulation, the surface transmission must be considered for the cold side as well as for the warm side of such insulation. In such cases equation (5) is applicable. The thickness of insulation given by this equation is less than that given by equation (6) by k/c', which is equal to 0.24 in. when $k = 0.3$ and $c' = 1.25$. The flat surface curves in Fig. 3 may therefore be used to determine the insulation thickness necessary to prevent sweating in the case of such cold spaces, 0.24 in. being subtracted from the thickness given by the curves to obtain the final result. For conductivities other than 0.3 it will be noted from equation (5) that the thickness required will vary directly with the conductivity. The thickness of insulation may also be reduced according to the insulating effect of the other materials entering into the construction of the walls of such cold space.

USE OF THE CURVES

The following variables should be considered: temperature difference $(T'-T''')$ between air on the warm side of the insulation and the cold medium, relative humidity, pipe size, conductivity of insulation, and amount of air circulation. The maximum thickness of insulation will be required when the temperature difference $T°-T'''$ is the greatest, when the humidity is the highest, when the conductivity is the greatest, and when the air circulation on the warm side of the insulation is the least.

Corrections for conductivity and air circulation have already been discussed. Humidity is the greatest and the most important variable. The following observations may assist in determining the humidity to be allowed for, in structures under design. In the northeastern states the relative humidity of the outside air during the winter months varies generally between 30 and 100 per cent. The air within heated buildings

which are not equipped with any means of humidification will vary generally during these months from 15 to 35 per cent. Where the buildings are equipped with humidification systems for the purpose of maintaining bodily health and comfort the humidity will vary from 40 to 50 per cent. During the summer months also the humidity of the outside air will vary generally between 30 and 100 per cent, but the humidity inside of buildings will run slightly higher than that of the outside air, except in the case of factories where heat is being used or generated, where the humidity will be less. During fogs the air is saturated and the humidity is 100 per cent.

Where a structure is already in existence, the humidity therein can be determined at any given time by means of a wet and dry bulb thermometer. If desired, the dew point may be determined directly by placing a thermometer in a silvered glass or polished brass test tube which is half filled with a mixture of ether and alcohol. Agitate the mixture by blowing air into it through a rubber tube, and the thermometer reading when dew appears on the polished outer surface of the test tube will be the dew point.

CHOICE OF INSULATION

After arriving at the correct thickness to prevent sweating on the outer surface of the insulation the next important point is to choose an insulation that is as nearly air tight as possible. If any air reaches the cold surface through cracks or joints in the insulation moisture will be deposited on the surface. The insulation will then become gradually water soaked from the inside and rendered worthless as an insulator. Cork appears to be the most suitable material for the purpose, as it lends itself readily to the attainment of an air tight job. All joints and cracks can be effectively sealed with brine putty. When other materials are used, special effort should be employed to protect them from moisture and to obtain a minimum of air infiltration.

DISCUSSION

F. PAUL ANDERSON: The conscientious work of Mr. Barrett is of great interest and this subject on which there is so little data is a field of endeavor that the Research Laboratory should undertake very definitely at an early date. It is our desire to carry on work that will be of value to the Society and industry represented by it and I think that if Mr. Barrett will press the point we can perhaps go into the subject of pipes buried in the ground or this particular problem he has discussed this morning.

E. S. HALLETT: The effect of air motion on sweating was of particular interest to me as this is often a problem in school buildings particularly on Monday mornings after the heating plant has been shut down for the week end. This condition is not found in our schools with the all-blast system, apparently because of the air motion. Our experience agrees with Mr. Barrett's in the case of unventilated basements.

H. M. HART: I am very glad to compliment Mr. Barrett for his admirable paper as such papers are of great benefit to the members and tend to keep up interest in the Society and maintain its prestige. It also answers the question of some who have remarked that the members seem to be letting the Research Laboratory present all the papers of real value.

No. 662

FLOW OF HEAT IN BUILDINGS

By F. E. Giesecke[1], Austin, Texas

Member

SINCE the walls, floors, roofs, and other structural elements of buildings are not homogeneous throughout their extent, the thermal resistance of those elements varies from point to point, and in order to determine its average value it is necessary to use either a large number of thermocouples or a resistance wire of considerable length.

About two years ago the Lone Star Ice Co., Austin, Texas, began the erection of a cold storage building, Fig. 1, which contains a room about 40 x 50 x 24 ft., used for the storage of ice, where the inside temperature is about 30 deg. fahr., when the outside temperature is about 90 deg. fahr.

It was thought that this room would afford an exceptional opportunity for the study of the flow of heat into buildings and the owners kindly permitted the investigation.

First preliminary studies were made to determine the best method of measuring the flow of heat. The plan finally adopted was the use of 100 ft. of No. 20 rubber-covered fixture wire in the area in which the temperature was to be determined. Repeated trials proved that by this method the temperature could be determined accurately within 2 deg. fahr. by measuring the varying electrical resistance of the wire.

To eliminate the effect of that part of the wire which was not within the area in which the temperature was to be determined and which was, therefore, generally at a different temperature, two wires, differing from each other in length by 100 ft. were used in every case. The two wires were placed so that both wires extended well into the area in which the temperature was to be determined, and so that the portions of the two wires which were not in the area in which the temperature was to be measured were under like conditions regarding temperature influences.

The difference between the electrical resistances of the two wires at any one time was therefore the resistance of 100 ft. of wire at the

[1] Head, Engineering Research Division, University of Texas, Austin, Texas.

Paper presented at the Semi-Annual Meeting of the American Society of Heating and Ventilating Engineers, Chicago, Ill., May, 1923.

desired temperature. By calibrating the wire the resistances could be easily translated into temperatures. (This method was suggested by Dr. S. L. Brown of the Department of Physics.)

It was first thought necessary to calibrate each wire separately, but it was found that the wire was so uniform in its electrical resistance that the same calibration curve could be used for all wires tested except in a few cases where extreme accuracy was desired. The calibration curve for 100 ft. of No. 20 fixture wire is shown in Fig. 2.

As the construction of the building progressed, resistance wires were placed in the floor, walls, ceiling, and roof of the building. The terminals of the wires running into the ceiling, roof, and south wall may be seen in Fig. 3.

The construction of the building is shown quite clearly in Figs. 4, 5 and 6, which represent, respectively, sections through the floor, through the south wall, and through the roof.

To determine the flow of heat through the floor into the building, resistance wires were placed as shown in Figs. 4 and 7. Wire No. 1 is resting directly on the ground and under the lower concrete slab. Wires Nos. 2 and 3 are under the cork and above the lower concrete slab. Wire No. 4 is on the cork and under the upper concrete slab. Wire No. 5 was laid on the upper concrete slab, and pressed ¼ in. into the concrete.

The reason for using two wires below the corkboard was that it had been planned to use two layers of corkboard, each one 3 in. thick and it was intended to have wire No. 3 between the two layers of corkboard, but during construction it was found that the 6 in. layer of corkboard had to be built up partly of two 3-in. layers and partly of one 4-in. and one 2-in. layer, and for that reason it was decided to use both wires under the corkboard.

To determine the flow of heat through the south wall into the building, wire No. 6 was placed between a coat of cement plaster and the hollow tile, and wire No. 7 between the hollow tile and the 4-in. layer of corkboard as shown in Fig. 5.

To determine the flow of heat through the roof and ceiling Wire No. 8 was attached to the under side of the lower corkboard and covered with a coat of pitch so that it gave the temperature of the mastic separating the lower sheet of corkboard from the air in the upper part of the room. Wire No. 9 was placed between the two layers of corkboard and wire No. 10 was placed between the upper corkboard and the wooden ceiling. Wire No. 11 was placed on the wooden ceiling while wire No. 12 was attached to the under side of the rafters. Wire No. 13 was placed on the composition roof and covered with one layer of roofing felt. The construction of the roof and ceiling is shown in Fig. 6 but may be clearer from the following description:

The roof consists of ⅞ in. ship lap, supported by 2 x 6 in. rafters and covered with a four-ply Phoenix composition roof.

The ceiling consists of a layer of yellow pine boards having an average thickness of 2.02 in., not fitted together but separated slightly so that the area of the air spaces is about 0.0035 of the total ceiling area; and of two layers of corkboard, the upper layer being 2 in. thick and the lower layer 3 in. and having a ¼ in. mastic finish.

The roof and ceiling are supported by combination wood and iron trusses; the average distance from ceiling to roof is about 5 ft. 6 in. The space between ceiling and roof is confined on all sides and since the ceiling is cooler than the roof there is no apparent cause for circulation of the inclosed air, and it would seem that this volume of air should be at rest and should have a fair thermal resistance.

FIG. 1. HEAT FLOW IN COLD STORAGE BUILDING STUDIED

Observations of the temperatures as indicated by the various resistance wires have been made at frequent intervals during the past twelve months.

The building was completed in July, 1922, and the cold storage room, which is under investigation, was put into use in August. The gradual cooling of the component parts of the floor is shown in Fig. 4 from which it appears that the temperature in the room has averaged about 30 deg. with a minimum of about 22 deg. in November and a maximum of about 33 deg. in May, and that the temperature of the upper part of the earth under the concrete floor fell rapidly from about 75 deg. in August, 1922, to about 60 deg. in February, 1923, and has remained practically constant at that temperature to the present time. When this floor was designed, experienced engineers expressed the fear that the ground under the floor would freeze and, as a result, expand and break the floor, but the temperature, as shown in Fig. 4, proves clearly that the thermal resistance of the floor as constructed is considerably greater than necessary to prevent freezing of the ground under the floor in the locality (Austin, Texas) where this building was erected.

When this investigation was begun, it was assumed that it would be comparatively easy to determine a uniform flow of heat through the inclosing members of the building, and consequently, a constant temperature gradient, and then by knowing the thermal resistance of one component part—corkboard, for example—the thermal resistance of the remaining members in that particular portion of the building could be calculated.

The investigation, however, proves that the flow of heat through the building is constantly varying and that it is therefore difficult to determine a uniform temperature gradient from which conclusions re-

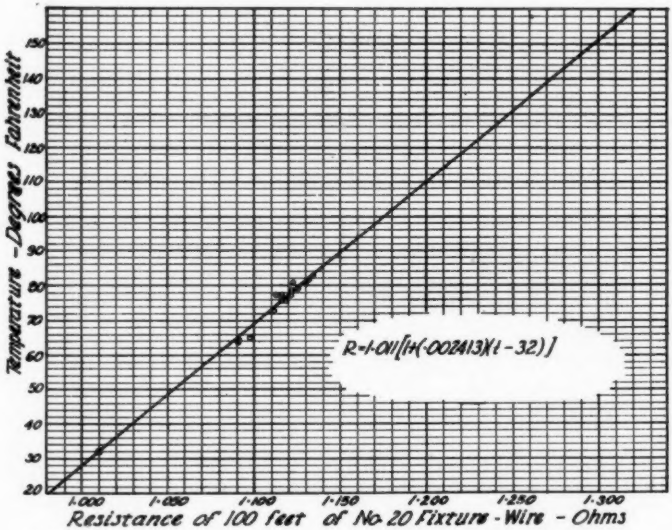


FIG. 2. CALIBRATION CURVE FOR 100 FT. OF FIXTURE WIRE

garding the thermal resistances of the component parts of the structure can be drawn.

However, during the month of February the temperatures were practically constant throughout the floor and the temperature gradient, as shown in Fig. 4, should represent quite accurately the relative thermal resistances of 4 in. of concrete, 6 in. of corkboard, and 6 in. of concrete. It will be noted that these resistances are to each other, respectively, as 5:16:9; in other words, that the thermal resistance of this particular corkboard is to the thermal resistance of this particular concrete slab as 16 is to 9.

Since this corkboard cost about $1.08 per cu. ft. and the concrete about $0.26 per cu. ft. the cost of the corkboard is a little more than twice that of concrete having the same insulating value. It appears, in other words, that for a floor of a cold storage building of the type

shown in Fig. 1, the same degree of insulation can be secured by the use of concrete as by the use of corkboard and at about one-half the cost.

This result came as a great surprise to the writer and can evidently only be explained by the probability that the corkboard is saturated with water and has a much lower thermal resistance than dry corkboard would have, and that the corkboard is somewhat compressed by the load of ice it supports, which exerts a pressure of about 1000 lbs. per sq. ft.

To determine the effect of such a pressure, a piece of corkboard, 12 in. sq. was subjected to a load of 1000 lbs. and it was found that the corkboard, when dry, was compressed about 0.12 in. in 6 in., and,

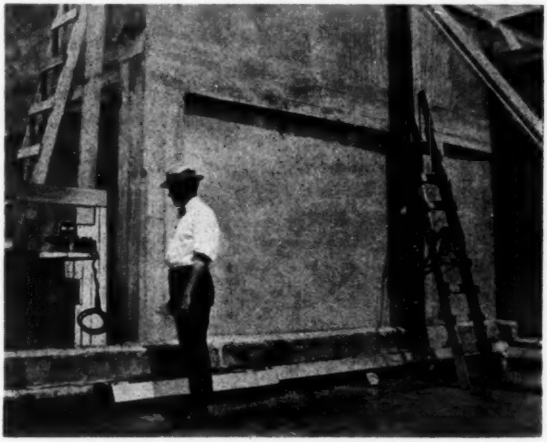


FIG. 3. RESISTANCE WIRES INSTALLED DURING CONSTRUCTION

when thoroughly wet, about 0.14 in. The increase in height due to the absorption of water was, however, slightly more than 0.02 in. in 6 in. so that the actual thickness of a layer of corkboard, nominally 6 in. thick and supporting a load of 1000 lbs. per sq. ft., is slightly greater when thoroughly wet than when dry.

When this investigation was begun, the writer assumed that the thermal resistance of corkboard was quite constant and accurately known and that he could use the corkboard as a standard by which to judge the thermal conductivity of the other materials used in the structure.

According to Professor Norton of the Massachusetts Institute of Technology, the thermal conductivity of corkboard is about 0.27, Nonpareil Corkboard Insulation.

According to Prof. A. C. Willard the thermal conductivity of corkboard is about 0.32 (Bulletin No. 102; 1917; University of Illinois).

According to M. S. Van Dusen, of the Bureau of Standards, the thermal conductivity of corkboard, having a density of 16, is 0.35. (JOURNAL AM. SOC. HEAT.-VENT. ENG., October, 1920.)

According to Dr. W. Nusselt, the thermal conductivity of corkboard having a density of 12.4 is 0.495 at a temperature of 64.4 deg., and, in general, the conductivity of insulating materials increases with temperature as indicated by the following expression:

$$c_t = c_{32} + 0.0006\ (t - 32)$$

(Zeitschrift des Vereines Deutscher Ingenieure, June 20, 1908.)

In order to determine the thermal conductivity of the particular corkboard which was used in this building and which had a density of 9.6, tests were made in the physical laboratory of the University of Texas, under the direction of Dr. S. L. Brown, and, according to these tests, the thermal conductivity varied from 0.35 to 0.6, some of the tests being with slabs and some with cylinders.

As a result of this investigation the writer believes that the thermal conductivity of corkboard varies with the moisture content, with the density of the corkboard, and with the temperature at which the test is conducted.

If we assumed that the conductivity of 1:5 concrete is 9.5 (Eng. Exp. Sta., University of Illinois, April 25, 1921), the conductivity of the corkboard in the floor under investigation must be 5.4 or about ten times as high as it should be for dry board.

That the conductivity of porous materials increases very rapidly with the moisture content of that material is shown very clearly by experiments conducted by Dr. Knoblauch (Gesundheits-Ingenieur, April 30, 1922), according to which the conductivity of brick increases from 3.3 to 6.6 when the brick is allowed to absorb 1.8 per cent (by volume) of water.

In the case of the corkboard under investigation, it was found that when the corkboard was saturated it contained 3 per cent (by volume) of water.

In studying the flow of heat through the roof and ceiling, quite a number of tests were conducted. Fig. 8 shows the results secured May 16, 1922. A casual inspection of this diagram shows:

1. That the temperature of the room was not increased by the higher temperature of the roof. The explanation of this is as follows: At the beginning of the test the air in the room was warmer than the lower layer of corkboard; it was about 4 p. m. before the temperature of the lower corkboard had reached that of the air in the room and hence the corkboard could not transmit heat to the air until after 4 p. m.; the test was ended at 5:45.

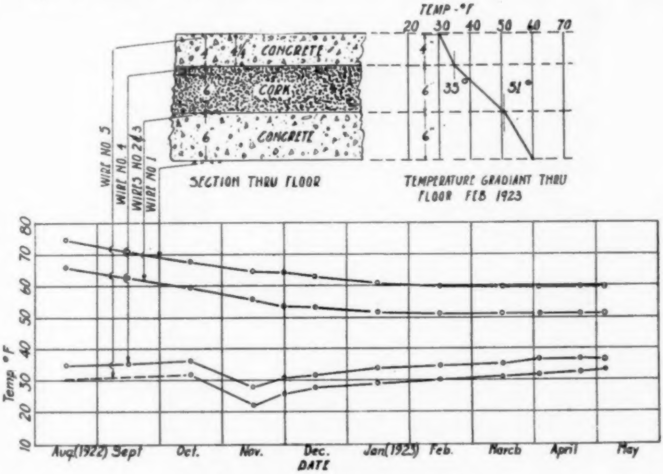
2. The thermal resistance of the 2 in. pine ceiling was not very different from that of the 2 in. corkboard underneath it because the distance between lines 11 and 10 between lines 10 and 9 do not differ materially from each other.

3. The thermal resistance of the 5½ ft. layer of air is very much less than that of the 2 in. wooden ceiling, because the distance between lines 12 and 11 is only about half the distance between lines 11 and 10, although more heat was passing through the air space than through the wooden ceiling.

4. The temperature shown by Wire No. 13 is only the temperature of the copper wire and of the uppermost portion of the composition roof; not that of

the upper ship-lap surface, because, if it were, the thermal resistance of the light roof construction would have to be very much higher than it can possibly be.

5. The temperature gradient through the roof and ceiling is continually varying and, as a consequence, it is difficult to determine the relative thermal resistances of the component parts of the roof and ceiling. It appears, however, that the temperature gradient remains fairly constant for a considerable portion of the afternoon and, for that reason, a number of readings were taken at that time of the day on various days with the results shown in Fig. 9. According to these results, the thermal resistance of the 5 ft. of air, of which the lower part is the cooler, is to that of 2 in. of wood as 16 is to 77; in other words, the insulating value of this space is less than that of a layer of wood one-half inch thick. This result also came as a surprise to the writer because he assumed that an air space in which there is no tendency for circulation should have a better insulating value than shown by this investigation.



Curves for Temperature in Floor of Ice Vault

FIG. 4. SECTION THROUGH FLOOR

The results of Fig. 9 also show that wood is not as good an insulator as corkboard, which agrees quite well with the generally accepted thermal resistances of those two materials.

In studying the flow of heat through the south wall, the greatest difficulty was experienced interpreting the results. Fig. 10 (Line 6) shows that the temperature of the outer portion of the hollow tile varied from about 90 deg. early in the morning to about 140 deg. in the middle of the afternoon; that the temperature of the hollow tile in contact with the corkboard (Line 7) varied from about 98 deg. early in the morning to about 92 deg. and then to about 108 deg. late in the afternoon, whereas the temperature of the outside air as shown by the dotted line in Fig. 10 never exceeded 100 deg. This shows that during the entire day the temperature in the tile wall was considerably higher than the temperature of the outside air, and that, therefore, the heat passing through the wall into the room is much more than

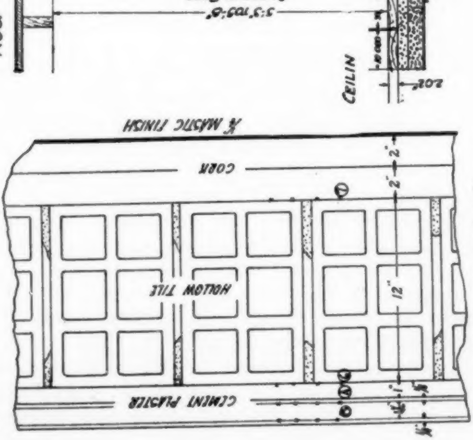


FIG. 5. SECTION OF WALL

FIG. 6. SECTION THROUGH CEILING AND ROOF

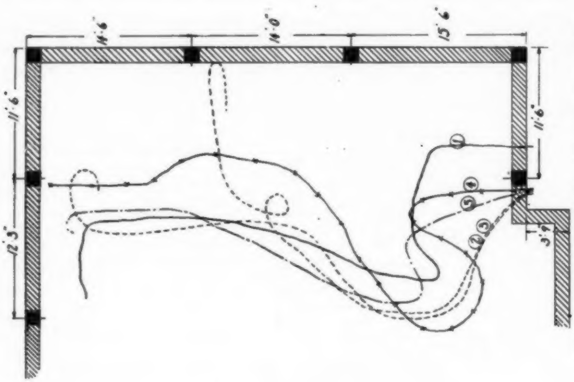


FIG. 7. POSITION OF WIRES FOR MEASURING HEAT FLOW

that which is calculated on the assumption that the outside air is warmer than the wall.

Since, with these results, it is practically impossible to determine the thermal resistance of hollow tile as laid in a wall, it was decided to make the determination when the wall was protected from the direct rays of the sun. To do this, a scaffold was built and heavy canvas stretched in front of the wall. In order to determine the heat flow more accurately, an additional layer of cement plaster was applied to the outside of the wall and two additional sets of resistance wires, as shown in Fig. 5, were installed. It was hoped that the thermal resistance of this plaster could be determined accurately and in that way, the heat flow into the building studied with satisfactory results, but this was not the case.

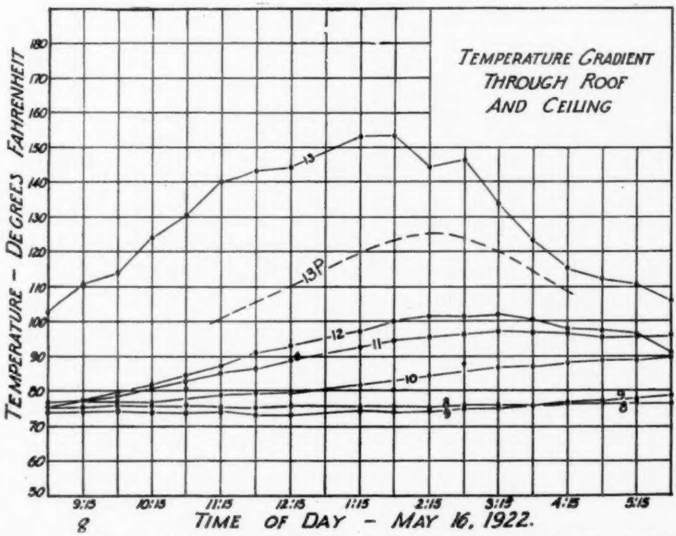


FIG. 8. RESULTS OF MAY 16, 1922, TESTS OF HEAT FLOW THROUGH CEILING AND ROOF

The wall absorbed heat throughout the day and the temperature of the hollow tile in the wall was higher than the temperature of the outside air; however, the increase in temperature was not as great as it was when the wall was exposed to the direct rays of the sun.

In order to determine the relative thermal resistance of the 12-in. hollow tile as compared with that of the 4-in. corkboard as shown in Fig. 5 for the 4-hr. period from 2 to 6 o'clock, September 7, it is evident that (Fig. 10, Line 6) the outside temperature of the hollow tile varied between 132 deg. and 139 deg. with an average of about 134 deg. and that the average temperature of the inner surface of the

tile (Line 7) varied from 96 deg. to 108 deg. with an average temperature of about 102 deg., and that the inner surface of the corkboard was about 32 deg. while the outer surface had the same temperature as the inner surface of the hollow tile; hence the average temperature gradient through the tile and corkboard was from 134 deg. to 102 deg. and to 32 deg., and that, therefore, the thermal resistance of the hollow tile was to that of the corkboard as 32:70, or by reducing these values to equal thickness, the resistances were to each other approximately as 11:70. Hence, if it is assumed that the conductivity of the

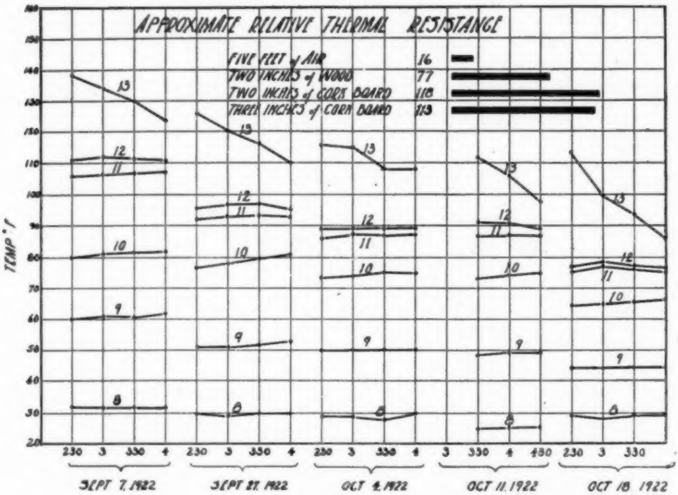


FIG. 9. STUDIES SHOWING RELATIVE INSULATION EFFICIENCIES OF WOOD AND CORK BOARD

corkboard is about 0.40, the conductivity of this particular type of hollow tile would be about 2.6.

This comparison is not quite accurate because the temperature of the joining faces of the hollow tile and corkboard increased from about 96 to about 108 deg., whereas the outer surface of the two materials remained constant as shown by the lines 6, 7 and 8 of Fig. 10; consequently, a small quantity of heat was stored in the tile and also in the corkboard during the four hour period, and therefore, the conductivity of the hollow tile is slightly greater than that indicated by the calculation above.

The conductivity of hollow tile indicated by this test is somewhat higher than that which the writer expected and he believes that it is desirable to do additional research work to determine the actual flow of heat through hollow tile walls of various types, when the inside (artificial) temperature is higher than the outside (natural) temperature, so as to be able to work with a fairly constant temperature

gradient, instead of with the constantly varying gradient existing in the walls of the building under discussion in this paper.

In conclusion, the author wishes to express his gratitude to Dr. J. M. Kuehne and Dr. L. S. Brown of the Department of Physics, and to Professors J. M. Bryant and J. W. Ramsey of the Department of Electrical Engineering, for their kind co-operation in this investigation, and particularly to Dr. L. S. Brown, who devoted much time to the general planning of the work and to the laboratory studies of the thermal resistance of corkboard and cement plaster; and also to W. M. Wilson, senior student in architectural engineering, who did the

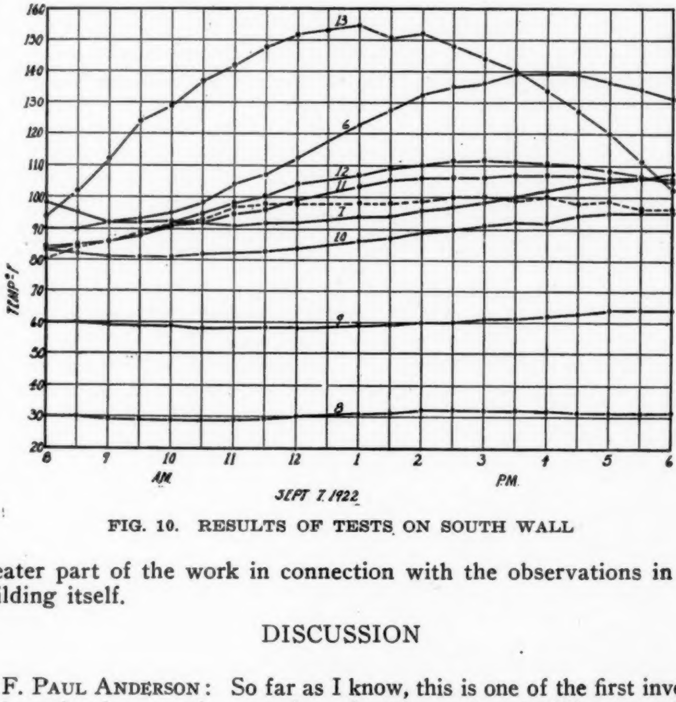


FIG. 10. RESULTS OF TESTS ON SOUTH WALL

greater part of the work in connection with the observations in the building itself.

DISCUSSION

F. PAUL ANDERSON: So far as I know, this is one of the first investigations that has ever been made on heat losses in a building complete. Much has been done in reference to the heat losses in various types of building construction on heat transmission through materials of various sorts. This, to my opinion, is one of the most admirable presentations on this subject that has ever been given to the engineering fraternity. Mr. Giesecke has been practically alone in carrying on investigations for a number of years on subjects that are very valuable to the heating and ventilating engineers and this is another illustration of the enthusiastic, painstaking methods that he employs in getting results.

JOHN G. SHODRON: About a year ago I had occasion to do work similar to that of Professor Giesecke, but in place of measuring the drop of temperatures by the resistance of wire, I used thermocouples. I worked in a room about 18 ft. wide, 7 ft. high and 56 ft. long, in which I had over 1000 thermocouples. I had thermocouples below the floor, in various layers in the floor, on the floor surface, the same with the walls, the underside of the ceiling and on the underside of the roofing and on the roofing surface. I was very much surprised to find the work that we did last winter is corroborated by the work that Professor Giesecke did; especially so was that of the conduct through the floor. I had occasion there, to try cork and the results indicated little advantage in using it. We took the floor off the ground, building it about 18 in. above the ground and then we worked with an air space of 3¾ in. thickness and got considerably more insulating value than we obtained from the 2 in. of cork. We tried the same thing on the ceiling, but the conductivity value of the dead air space in the ceiling didn't give us as big a drop in the temperature gradient. We laid that to the heat, that is, convection currents, the heat being applied on the bottom side of the ceiling and the dead air space transferring it from the ¾ in. ceiling to the 3¾ in. of dead air to the underside of the room. I would like to see considerably more work done in this line because I believe it will mean much to the engineering profession.

L. A. HARDING: Corkboard is the standard insulation, of course, in this country, and in practically all countries, for cold storage work. I have run a great many tests myself years ago on the apparent value of corkboard and practically all the published results so far have shown that 1 in. of corkboard, roughly, is just as good as a 13 in. brick wall. That of air space, of course, is largely dependent upon how well it is blocked off.

JOHN G. SHODRON: There was a 2-in. bed of sand placed on top of the natural soil and a 3-in. concrete slab, 2 in. of cork and a floor surface 2 in. thick. The temperature on the underside of the concrete floor during three weeks averaged about 45 deg. The temperature on the top side of the floor averaged about 51 deg. We built a second floor 18 in. above the soil, sheeted off the underside of the floor joists with ⅞ in. boards and covered them with 2-ply roofing. Then we had 3¾ in. air space on top of the 2 x 4 joists and we had ¾-in. finished wood floor. We stopped the infiltration from underneath the floor to the floor by means of the roofing paper and the temperature went down below or around 45 deg. We were able to maintain a considerably higher floor temperature than we could with the floor that was laid directly on top of the soil. The only thing that we could lay this thing to was the air space, the conduct value of the air and that of the dead air space underneath the floor.

PROFESSOR GIESECKE: Perhaps I should say that before we began this investigation several persons advised the use of thermocouples rather than resistance wires, but I believe that for this particular work resistance wires are better than thermocouples.

No. 663

HEAT TRANSFERENCE AND COMBUSTION TESTS IN SMALL DOMESTIC BOILER

By John Blizard[1] (Member)

and

W. M. Myler, Jr.[2], J. K. Seabright[3], C. P. Yagloglou[4] (Non-Members)

Pittsburgh, Pa.

Introduction

The Fuel Section of the U. S. Bureau of Mines is co-operating with the Research Laboratory of the American Society of Heating and Ventilating Engineers at the Pittsburgh Experiment Station of the Bureau in an investigation of combustion and heat transference in a small domestic boiler, and this paper gives an account of the experiments so far carried out.

The investigation was planned with the view of determining the relative amounts of heat absorbed by the fire-pot and each of the sections above it, the amount of free oxygen which finds its way from the ash-pit to the space above the fuel-bed, and the value of the ordinary methods of admitting air through the fire-door to burn the combustible gases rising from the fuel-bed. The method decided upon was to test first of all the fire-pot alone so arranged that no secondary air could be admitted over the fuel-bed, then to add the various sections to the fire-pot and carry out successive tests to see how the efficiency increased or decreased with the addition of each section, with and without the admission of secondary air through the ports in the fire-door. Incidentally, it was decided to place the boiler on scales to determine the efficiency and heat losses with an accuracy greater than that usually obtained, though no novelty is claimed for this idea.

The principal function of a domestic boiler is to maintain a house at a comfortable temperature with as little expense and attention as possible. The principal factors determining the ability of the boiler

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to heat a house, provided that sufficient amount of radiator surface is furnished, are the rate of combustion, which may be maintained with the draft available, and the efficiency. The frequency of attention required depends mainly on the quantity of fuel which may be placed in the fire-pot, the ash content and other qualities of the fuel, and on the efficiency. Obviously, the higher the thermal efficiency the less will be the required rate of combustion for the same output, and the longer will the fuel charged last. Now the thermal efficiency for any particular rate of steaming may usually be raised by increasing the number of sections up to a fixed number, which must be determined by experiment; but the number of sections which may be employed is limited by the cost of the boiler, its height, and the extent to which the draft would be reduced as sections are added. There is also the possibility of actually decreasing the efficiency by adding a section, for the added section may actually lose more heat by radiation and convection than is imparted to it by the gas in passing over it. At low rates of output with the ordinary boiler, the top section could generally be removed and the efficiency raised; but were it removed the efficiency at the higher ratings would be lower than it was before it was removed. In designing the most economical boiler the number of sections should thus be chosen so that the efficiency will be highest at the rating at which the greatest quantity of fuel is burned throughout the season.

FIG. 1. SECTIONAL VIEW OF THE COMPLETELY ASSEMBLED BOILER

It is important to bear in mind that each successive section added to a boiler is of less value than the section preceding it, and it is for this reason that the heating surface of a boiler is not an exact measure of its capacity. Thus, suppose a boiler consisting of a fire-pot and dome alone, with a heating surface of 10 sq. ft., to be generating steam at a pressure of about 2 lb. per sq. in. above that of the atmosphere, with a flue-gas temperature of 840 deg. fahr., a dry flue-gas loss of 25 per cent, and a thermal efficiency of 58 per cent. Then, if we postulate that each section with 3 sq. ft. of heating surface added to the boiler will reduce the temperature of the flue-gases by ½ deg. per mean degree temperature difference of the gas and the water, the results in Table 1 will be obtained:

TABLE 1. EFFECT OF ADDING SECTIONS TO VERTICAL HEATING BOILER

BOILER	Flue-gas Temp. Deg. fahr.	Drop in temp. per Section Deg. fahr.	Total Heating Surface Sq. Ft.	Thermal Efficiency Per Cent	Efficiency as per cent of maximum attainable	Efficiency as per cent of that with fire-pot and dome
Fire-pot and dome	840		10	58.0	74	100
1 section added	596	244	13	65.9	84	114
2 sections added	448	148	16	70.7	90	122
3 sections added	358	90	19	73.7	94	127
4 sections added	304	54	22	75.4	97	130
5 sections added	271	33	25	76.5	98	132
6 sections added	251	20	28	77.1	99	133
1000 sections added	220	0	3010	78.1	100	135

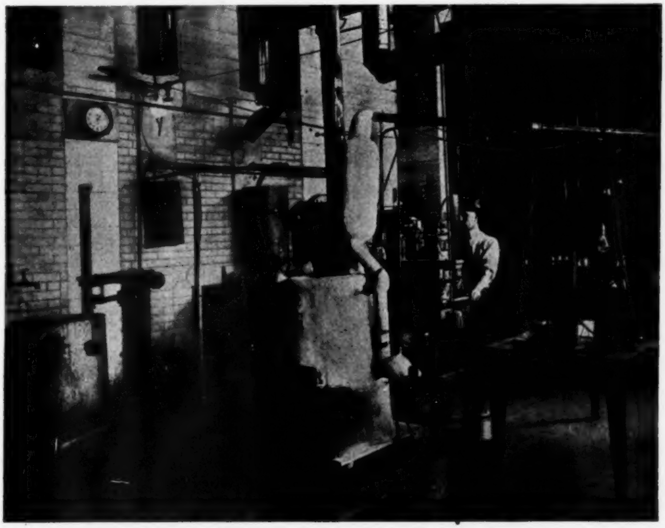


FIG. 2. FIRE-POT AND BASE OF BOILER ON SCALES FOR TESTING—FIRE-DOOR SEALED, FUEL BEING FIRED THROUGH COVER OVER FIRE-POT

Here it will be seen that doubling the heating surface will increase the output, which, for the same rate of combustion, is proportional to the thermal efficiency, by less than 30 per cent. And, while adding the first section to the fire-pot increases the efficiency and output by 14 per cent of its original value, adding the fifth section increases the efficiency and output by only about 1½ per cent of its value for the fire-pot and four sections. Actually, the gain in thermal efficiency and output would be less than given before, for the radiation and convection losses, and air infiltration, would increase with each section, so that it is doubtful whether the thermal efficiency would actually increase at all on adding the fifth section for the particular rate of combustion to which the above figures refer, and as the rate of combus-

tion decreased, so would the number of sections which would give the best efficiency decrease.

All of the above figures, with the exception of those for the fire-pot and dome, are imaginary, but they show the kind of information the trials will give when completed. They will include also other information of vital importance in boiler design, such as the effect of additional sections on the draft, and the effect of the size and number of holes in each section for the gas passages on heat transmission, eventually; a boiler of a somewhat larger grate area may be tested to see how the heat transmission and composition of gases over the fuel-bed varies with the size of the boiler.

Not the least important part of the program is the determination of the difference in efficiencies with bituminous coal, coke, and anthracite, as the two former fuels are gradually taking the place of anthracite, which has so long been the standard fuel for small boilers. The Bureau of Mines, alive to this situation, has already carried out an important series of tests[6] on four domestic boilers for the comparison of these fuels, but they were not conducted in the elaborate manner laid out in the present program, which it is hoped may lead to the design of a boiler that will raise the efficiency when burning high volatile bituminous coal to somewhere near that obtained when burning anthracite or coke. There is ample scope for this improvement for the publication referred to shows that while the efficiency with coke and anthracite is about 70 per cent in an ordinary boiler, the corresponding efficiency with Pittsburgh coal was only about 54 per cent.

OUTLINE OF PRINCIPAL RESULTS

The trials so far carried out may be divided into two series—namely, (1) those in which the fire-pot and base only were used; and (2) those in which the fire-pot, base, and dome were used. The first series of tests was run with no secondary air supply, the fire-door being sealed up tightly and coal fired through an opening in the plate covering the fire-pot, and this opening being closed after firing so as to make it air-tight. During one-half of the second series, secondary air was admitted continuously through the slots in the fire-door provided for the purpose and during the other half no secondary air was admitted. Each series of trials was carried out with anthracite, coke, bituminous coal, and natural gas as fuels.

The results of these trials are shown in detail by means of tables and charts later in this paper; but the more important features are discussed in the following summary.

PASSAGE OF FREE OXYGEN PAST FUEL BED

The method of carrying out the first series of trials, whereby no air could reach the space over the bed of fuel except by passing

[6] Comparative Tests of By-Product Coke and other Fuels for House-Heating Boilers: By Henry Kreisinger, John Blizard, H. W. Jarrett, and J. J. McKitterick, Bureau of Mines Tech. Paper 315; to be published shortly.

through the grate-bars, made it possible to determine whether sufficient air can be supplied beneath the grate-bars of a small boiler to burn completely the fuel to carbon dioxide and aqueous vapor. Since the publication[6] of the results of tests on the combustion of fuel at low rates in a small furnace lined with firebrick, it has been assumed by many that very little oxygen could pass up from the ash-pit to the space over a homogeneous fuel-bed without combining with the fuel to form carbon monoxide, carbon dioxide, and water vapor; also

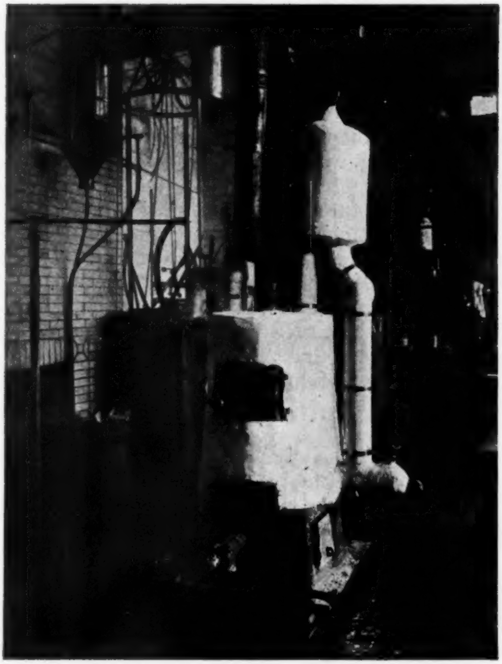


FIG. 3. BOILER COMPOSED OF DOME, FIRE-POT, AND BASE, EQUIPPED FOR BURNING GAS

that it is essential to admit air over the fuel-bed if considerable losses due to combustible gases, vapors, and soot are to be avoided, and the thermal efficiency is to be maintained reasonably high.

This assumption is not an unreasonable one; for the tests described in that publication showed that the fuel-bed of a furnace acts as a gas producer, that the oxygen is all used up as it passes through the first 3 or 4 in. of the fuel-bed, and that the gases rising from the top of the fuel-bed contain large quantities of carbon monoxide, and some hydrogen, hydrocarbons, and soot, depending upon the constitution of the fuel. But these results are based on the determination of the

composition of the gas in the fuel-bed itself, and while, therefore, they proved conclusively that any oxygen passing vertically upwards through the central portion of a homogeneous fuel-bed of an ordinary domestic boiler will soon be used up and form a large proportion of carbon monoxide, they do not permit us to assume that no oxygen can pass up near the outer water-cooled boundary of the fuel-bed.

The results of the first series of trials show conclusively that free oxygen finds its way from the ash-pit to the stack, and consequently that with anthracite and coke, the heat lost by not burning carbon monoxide was never over 5 per cent of the calorific value of the fuel. But when burning Pittsburgh coal the losses caused by combustible gases, vapors and soot passing up the stack were fairly high. These higher losses with Pittsburgh coal cannot be ascribed entirely to a lack of oxygen, because throughout any trial seldom was there insufficient oxygen to burn the combustible gases, vapors and soot. Evidently, since free oxygen was always present in the flue-gases, inadequate provision is made for igniting, mixing and burning the combustibles rising from the fuel-bed of the small furnace. One of the dominating factors militating against this is the low temperature prevailing above the fuel-bed, as during the first series of trials, the temperature of the products of combustion leaving the fire-pot was as low as 400 to 500 deg. fahr. at the lowest rates of combustion, and rose only to 950 to 1150 deg. fahr. at the highest rates of combustion, which latter temperatures are less than the ignition temperature of carbon monoxide (about 1220 deg. fahr.) and methane (1200 to 1380 deg. fahr.) and only about that of hydrogen (approximately 1080 deg. fahr.).

It is thus obvious that the gases and air must be ignited by some hot exposed portion of the fuel-bed or mixed with air which is preheated in some way, or some means taken to partially reduce the cooling action of the fire-pot if the combustible gases are to be burned. That raising the temperature of the reactants is an important factor in accelerating combustion is well known, and it was not surprising, therefore, to find in the trials that as the rate of combustion and with it the temperatures of the products of combustion increased, the excess air in the flue-gases decreased, though this in part may be ascribed to the higher velocities of the gases which caused them to mix more thoroughly.

SECONDARY AIR

Although at practically all times during the first series of trials sufficient air was present over the fuel-bed to burn the combustible gases, it was necessary to determine whether admitting further air over the fuel-bed would reduce the losses caused by the combustible gases. Accordingly, on the second series of trials, during the whole of one set, air was admitted through ports in the fire-door provided for the purpose, so that the results might be compared with another set during which the air-ports were closed. As might have been anticipated from the comparatively small carbon monoxide content of

the gases in the first series of trials when burning coke and anthracite, opening the air-ports during the second series when burning these fuels gave an efficiency lower than when they were closed. But with bituminous coal, the admission of secondary air raised the efficiency somewhat at the highest rating, though at the lowest rating the efficiency was lower when admitting secondary air. This distinction between the highest and lowest rates is not surprising, since at the lower rate a larger proportion of excess air enters from beneath the grates than at the higher rates, and, besides, the temperatures are higher at the higher rates.

In an attempt to reduce the combustible losses, the fuel was fired by the coking method, hoping thereby that the volatile matter might be driven off from the coal more gradually and be ignited by the hot

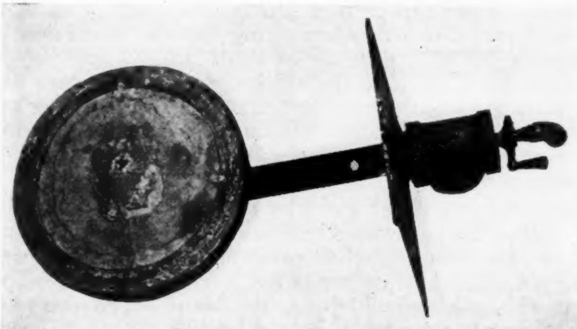


FIG. 4. GAS BURNER USED FOR TESTS WITH NATURAL GAS

coke which was pulled forward to make room for the fresh coal charge in the rear. In general, if the hot coke is left exposed, this method of firing works well; but this boiler is so small that after putting the full charge of fresh fuel into it the hot coke is covered entirely. And, while the freshly charged fuel above the hot coke burnt through more rapidly than when firing by the spreading method, the coking method of firing did not raise the efficiency. Undoubtedly the efficiency might have been raised had the air-ports been adjusted on the completion of each gas analysis to suit the secondary air requirements suggested by that analysis, but this procedure would have had no practical significance. The trials do show, however, that after firing, and particularly after stirring the fuel-bed, that secondary air could probably be admitted to advantage. The effect of admitting secondary air for a time after attending to the furnace may be examined in a later series of trials.

* Low-rate Combustion in Fuel Beds of Hand-Fired Furnaces: By Henry Kreisinger, C. E. Augustine, and S. H. Katz. Bureau of Mines Tech. Paper 139, 1918.

THERMAL EFFICIENCY

The overall thermal efficiency of this boiler when burning anthracite and coke varied between 51 and 59 per cent with the fire-pot alone as the heat-absorbing surface, between 57 and 60 per cent when the dome was added to the fire-pot and the ports in the fire-door were kept closed, and between 49 and 58 per cent when the ports were open. These efficiencies are surprisingly high for so elementary a boiler, and if the radiation and convection loss from the boiler can be regarded as useful heat, which is often justified, since it keeps warm the basement of a house, the total useful efficiency with coke and anthracite was about 70 per cent.

With bituminous coal the efficiency was much lower. It varied between 33 and 40 per cent with the fire-pot only, between 41 and 48 per cent with the fire-pot and dome and no secondary air, and between 40 and 44 per cent with a supply of secondary air. All these efficiencies were obtained when firing by the spreading method. When, during the second series of trials, the bituminous coal was fired by the coking method, the efficiency varied between 38 and 43 per cent with no secondary air, and between 41 and 46 per cent with secondary air. Thus there is a considerable scope for improvement in the design of the ordinary domestic boiler when burning a high-volatile bituminous coal.

When burning natural gas the efficiency, based on the gross calorific value of the gas, varied between 40 and 57 per cent for the fire-pot only and between 55 and 63 per cent when the dome was added to the fire-pot. The efficiency was not, therefore, on the whole, quite as good as with coke and anthracite, but this is largely due to the comparatively high loss caused by the total heat of the steam formed from the gas which left the boiler with the flue-gases.

DETAILED REPORT OF TRIALS

DESCRIPTION OF BOILER

The boiler used in these trials is a small cast-iron boiler and was given to the Bureau of Mines by the United States Radiator Corporation. The following are its principal dimensions:

Base:

Height, in.	13
Size of draft door, in.	6¾ x 6
Area through draft door, sq. in.	40½

Grate (rotary duplex):

Diameter, in.	20
Area, sq. ft.	2.2
Ratio air space to total area, per cent	55
Mean height above floor, in.	9¾

Fire-pot or first section:

Inside diameter (at top), in.	13½
Inside diameter (at bottom), in.	20½

Height (vertical), in.	21¾
Vertical distance, bottom of fire-door to top of base, in.	16
Fire-door opening, in.	8½ x 11½
Number of slots in fire-door for admitting secondary air	4
Total area through slots in fire-door, sq. in.	3.9
Clinker door opening, in.	3½ x 11¼
Heating surface, sq. ft.	7.6

Dome:

Diameter, in.	17¼
Diameter of central opening, to flue in.	6¾
Length of central opening, in.	7¾
Number of water connections	2
Diameter of water connections, in.	4½
Heating surface, sq. ft.	2.3

NOTE:—In groups of 3 curves show thermal efficiency curve: Points (△) relate to upper curve; Points (x) relate to middle curve; Points (o) relate to lower curve.

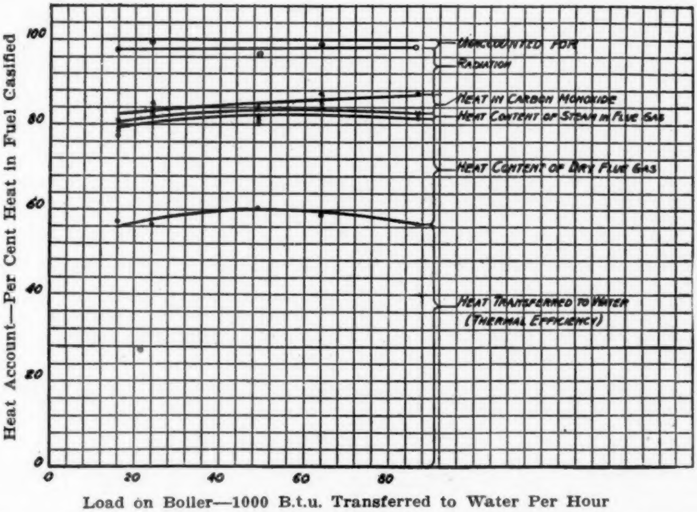


FIG. 5. HEAT ACCOUNT BASED ON HEAT IN FUEL GASIFIED. FIRST SERIES OF TRIALS WITH COKE, FIRE-POT ONLY, NO SECONDARY AIR

Fig. 1 shows a sectional view of the completely assembled boiler. A photograph of the fire-pot and base, set up on scales for testing, is shown in Fig. 2.

In Fig. 2 may be seen a rectangular door on the side which may be opened to the extent required to admit air to the lower side of the grates. Above the boiler is a steam separator so arranged that any water separated from the steam will return automatically to the boiler. The loss by radiation and convection from this separator was not inconsiderable, and decreased to some extent the efficiency of the boiler as determined by the trials.

NOTE:—In groups of 3 curves show thermal efficiency curve: Points (△) relate to upper curve; Points (x) relate to middle curve; Points (o) relate to lower curve.

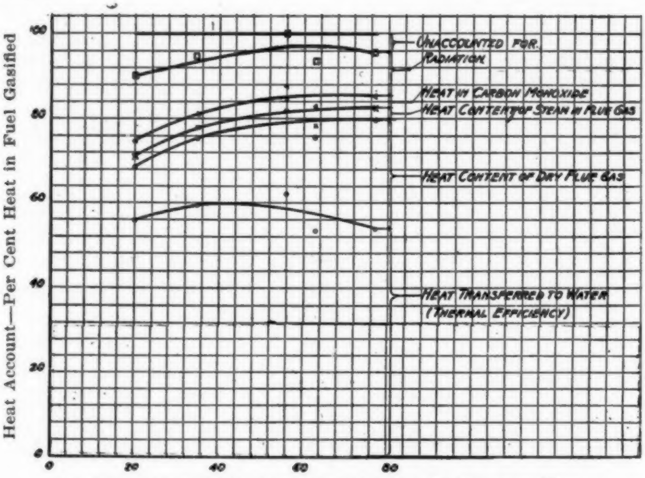


FIG. 6. HEAT ACCOUNT BASED ON HEAT IN FUEL GASIFIED. FIRST SERIES OF TRIALS WITH ANTHRACITE, FIRE-POT ONLY, NO SECONDARY AIR

Fig. 2 shows how the fire-door opening was completely sealed during the first series of trials. A steel plate, ¼-in. thick, covered the top of the

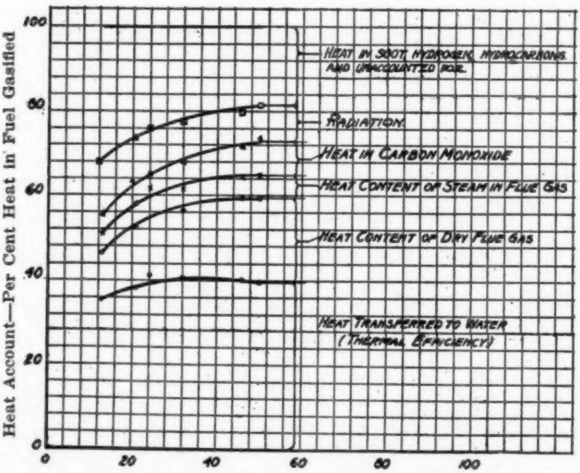


FIG. 7. HEAT ACCOUNT BASED ON HEAT IN FUEL GASIFIED. FIRST SERIES OF TRIALS WITH BITUMINOUS COAL, FIRE-POT ONLY, NO SECONDARY AIR

fire-pot, and in it was cut one opening through which to charge the fuel, another through which the gases passed to the stack, and two others through which the steam passed to the steam separator. The opening through which the fuel was charged was provided with a cover and asbestos gasket so that this opening might be sealed tightly after firing. On all trials, the outer surface of the boiler, separator, and connections, was covered with 1½ in. of asbestos insulation.

Fig. 3 shows the boiler as set-up for the second series of tests, and here the fire-pot may be seen with the dome only above it. When the photograph was taken, the boiler was equipped for burning natural gas,

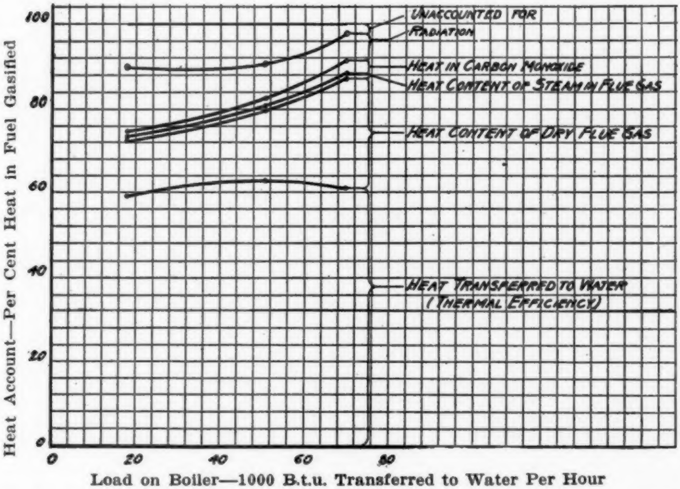


FIG. 8. HEAT ACCOUNT BASED ON HEAT IN FUEL GASIFIED. COKE, SECOND SERIES, FIRE-POT AND DOME, NO SECONDARY AIR, PORTS IN FIRE-DOOR CLOSED

and the gas-pipes may be seen connecting up to the regulator valve, and device for mixing the gas with a supply of primary air. It will be noticed that the steam separator shown in Fig. 3 is larger than that shown in Fig. 2; it was found necessary, owing to excessive priming, to use a larger separator.

Fig. 4 shows the gas burner used. The gas entered the *T* at the right, which served as a mixer. The burner proper consisted of a heavy sheet-iron ring of square cross-section. The gas escaped through 200 holes, 1/16 in. in diameter, arranged in two concentric circles with equal intervals between the holes. It was found necessary to cover up the central opening in the ring with a metal plate to obtain a steady flame. The burner was designed to burn enough gas to evaporate steam at the highest rates obtained with coal, and yet have a fairly short flame. At the lowest rates it was found necessary to plug-up about two-thirds of the holes in order to make the gas burn steadily.

METHOD OF CONDUCTING TRIALS AND COMPUTING RESULTS

The boiler was mounted on scales, and the gauge-glass was calibrated to show the weight of the water in the boiler. It was thus possible to weigh the exact amount of fuel and ash in the fire-pot at any time, and so determine the rate of combustion during any part of the trial, also the total weight of fuel gasified during the whole trial.

Approximately 200 lb. of fuel was fired on each trial. Each charge weighed 30 lb., with the exception of the last one, and was the same for all rates and all fuels. This weight of fuel fired at each firing was

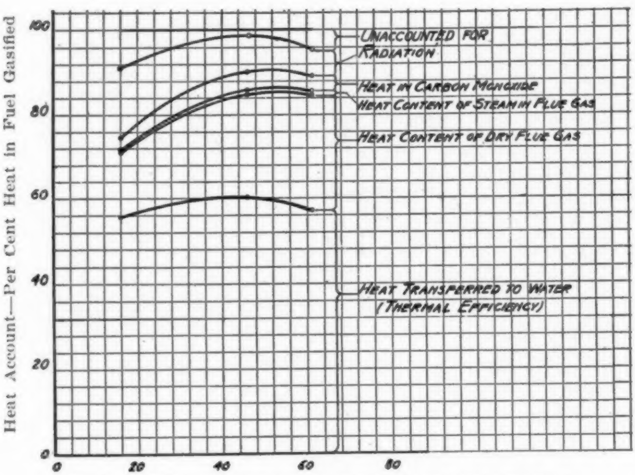


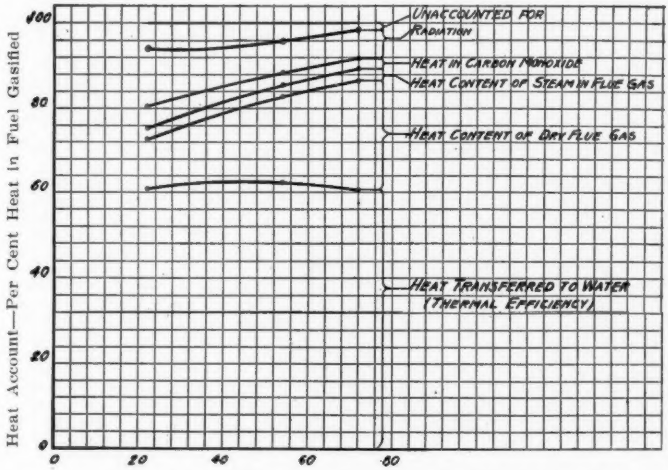
FIG. 9. HEAT ACCOUNT BASED ON HEAT IN FUEL GASIFIED. COKE, SECOND SERIES, FIRE-POT AND DOME, SECONDARY AIR PORTS IN FIRE-DOOR WIDE OPEN

chosen because it was the greatest amount of coke (the bulkiest fuel used) that could be fired at one time, and it was desired to fire all fuels at all rates in a precisely similar manner. The interval between firings was thus inversely proportional to the rate of steaming, and varied from 3 to 15 hr.

The rate of combustion was kept as nearly constant as possible throughout the trial by manipulating the draft, removing ash, and when necessary, breaking up the bituminous fuel-bed. As little attention as possible was given the fuel-bed. Fuel was charged when the weight of the boiler indicated that the previous charge had been consumed. No attempt was made at any time during the test to manipulate the fire or the draft dampers to improve the efficiency. The sole attention given, therefore, was that required to maintain a constant rate of evaporation.

From the total weights of fuel fired and refuse removed, the change in the weight of the contents of the fire-pot during the trial, and the com-

position and calorific value of the fuel charged and refuse removed, the total heat in the fuel gasified was calculated. It was assumed that all the ash in the fuel charged was either removed with the refuse from the ash-pit or remained in the fire-pot and that the fire-pot contained only ash and carbon. The first assumption, that all the ash in the coal was removed with the refuse or remained in the fire-pot, is not strictly accurate, since it is well known that some particles of ash are always carried up the flue with the flue-gases; but it is unlikely that, at low rates of combustion, the quantity of ash carried up the stack is large enough to cause the method used to be greatly in error. The latter assumption was probably



Load on Boiler—1000 B.t.u. Transferred to Water Per Hour

FIG. 10. HEAT ACCOUNT BASED ON HEAT IN FUEL GASIFIED. ANTHRACITE, SECOND SERIES, FIRE-POT AND DOME, NO SECONDARY AIR, PORTS IN FIRE-DOOR CLOSED

approximately true, since the trial was started and stopped just before it became necessary to charge the fuel, so that the bulk of the volatile contents of the fuel was driven out of the fire-pot.

The exact procedure in calculating the fuel gasified is here illustrated by a concrete example from Trial 4*A*, as follows:

During this trial 200 lb. of coal containing 11.9 per cent of ash, was charged, and 12.2 per cent of the weight of the fuel was removed as refuse which contained 84.6 per cent ash. Thus, 200 x 0.119, or 23.8 lb. of ash, was charged, 200 x 0.122 x 0.846, or 20.6 lb. of ash was removed, and the fire-pot contained 23.8 — 20.6, or 3.2 lb. more ash at the end than at the beginning of the trial. The weight of the contents of the fire-pot during the trial decreased from 36 to 35 lb., or by 1 lb., and since it contained 3.2 lb. more ash at the end than at the beginning, it must have lost 3.2 + 1.0, or 4.2 lb. of combustible during the test. (See item

11 Table 2.) Thus the total heat available for use was the sum of the total heat contents of the coal charged, and of the 4.2 lb. depletion of the combustible contents of the fire-pot. Assuming the 4.2 lb. of combustible to be carbon having a calorific value of 14,540 B.t.u. per lb., then since the coal charged has a calorific value of 12,573 B.t.u. per lb., the total heat available for use during the trial was $200 \times 12573 + 4.2 \times 14540$ B.t.u. or

$$\left(1 + \frac{4.2 \times 14540}{200 \times 12573}\right) \times 100 \text{ per cent of the heat in the fuel charged,}$$

which is shown in item 36 to be 102.4 per cent. By similar reasoning the heat in the combustible in the refuse removed may be shown to represent

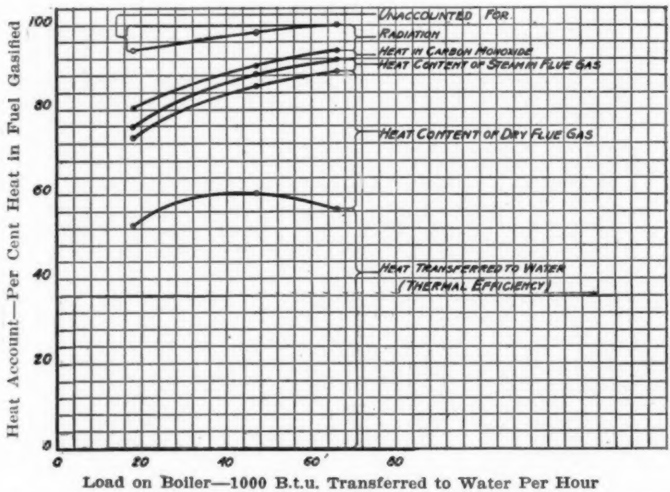


FIG. 11. HEAT ACCOUNT BASED ON HEAT IN FUEL GASIFIED. ANTHRACITE, SECOND SERIES, FIRE-POT AND DOME, SECONDARY AIR PORTS IN FIRE-DOOR WIDE OPEN

2.2 per cent of the heat in the coal charged; thus the total heat in the fuel gasified represents 102.4 — 2.2, or 100.2 per cent of the heat in the fuel actually charged. (See Item 38.)

The flue-gas temperature was measured by means of an exposed copper-constantan thermocouple and potentiometer. The hot junction of the thermocouple was placed in the center of the flue leading from the boiler and about 4 in. above the point where the flue joined the boiler.

The sample of flue-gas for analysis was drawn from a point near the hot junction of the thermocouple. It was drawn into a bottle at a steady rate for about a half-hour, and then analyzed in an ordinary Orsat apparatus. In some trials additional samples of gas were analyzed to determine the hydrogen methane, tar, and soot contents.

The draft was measured in the stack about 4 in. above the boiler, and in the ash-pit by means of an inclined U-tube filled with gasoline.

The water fed to the boiler was weighed, and its temperature was

TABLE 2. RESULTS OF FIRST SERIES OF TRIALS, FIRE-POT ONLY, WITH COAL AND COKE, NO AIR ADMITTED OVER FIRE-BED

Item No.	Particulars of observations	Units	Coke					Anthracite					Bituminous						Units
													Spreading					Coking	
1	Number of trial		3	2	4A	5	22	10	6	7	8	9	14	11	12	13	15	16	
2	Duration of trial	Hours	91.0	48.5	31	22	34	67	47	30	24	18	48.5	42.2	33.5	22.5	20.5	44	Hours
	FUEL																		
3	Calorific value per lb. as fired	B.t.u.	12,881	12,539	12,873	12,671	11,994	12,867	12,941	13,223	12,956	13,259	13,050	13,382	13,276	12,995	13,145	13,143	B.t.u.
	Proximate analysis as fired																		
4	Moisture	Per cent	0.9	0.4	1.0	0.7	2.0	1.9	2.6	2.8	1.2	1.0	1.9	1.8	2.2	2.3	2.3	2.1	Per cent
5	Volatile matter	" "	2.1	3.7	2.3	3.3	3.8	5.9	5.4	6.3	6.2	5.6	37.6	39.4	39.3	37.4	37.8	36.6	" "
6	Fixed carbon	" "	85.8	82.7	84.8	84.8	79.9	80.2	80.5	78.6	79.9	83.2	51.0	50.5	50.5	50.1	50.6	52.7	" "
7	Ash	" "	11.2	13.2	11.9	11.2	14.3	12.0	11.5	12.3	12.7	10.2	9.5	8.3	8.0	10.2	9.3	8.6	" "
8	Ratio fixed carbon : volatile matter		40.1	22.4	36.4	25.4	21.1	13.7	14.8	12.6	13.0	14.9	1.4	1.3	1.3	1.3	1.3	1.4	
9	Total charged during trial	Lb.	201.5	209.0	200.0	202.0	200.0	200.0	200.0	200.0	200.0	200.0	200.0	200.0	200.0	200.0	200.0	200.0	Lb.
	FUEL BED																		
	Mass of fire-pot contents																		
10	Increase of mass during trial	Lb.	-8.0	-4.0	-1.0	9.0	3.0	-2.0	-11.0	-9.0	-16.0	-5.0	1.0	9.5	-8.0	0.0	2.0	3.0	Lb.
11	Increase of carbon contents of fire-pot	"	-10.9	-16.0	-4.2	6.5	-9.7	-6.4	-20.4	-16.2	-25.2	-3.4	-1.0	10.0	-2.2	-2.7	-1.9	5.0	"
12	Increase of ash contents	"	2.9	10.0	3.2	2.5	12.7	4.4	9.4	7.2	9.2	-1.6	2.0	-0.5	-5.7	2.7	3.9	-2.0	"
	Depth of fuel bed																		
13	At start	Inches	11	9	7	9	7	9	9	9	8	8	7	8	8	8	7	6	Inches
14	At finish	"	8	8	7	9	7	8	9	9	10	8	6	6	9	7	10	6	"
15	Mean depth, before firing	"	9.1	8.9	8.1	8.9	8.6	8.8	10.0	8.9	10.2	10.3	7.1	7.0	8.9	8.1	7.3	---	"
16	Mean depth, after firing	"	16.1	15.4	14.9	16.5	14.4	13.0	12.3	15.1	17.0	15.6	9.6	12.6	13.9	12.8	10.9	---	"
	ASH AND REFUSE																		
17	Carbonaceous matter in ash and refuse removed	Per cent	30.7	32.6	15.4	19.7	11.0	42.7	35.2	39.5	32.4	38.0	45.1	22.0	19.6	16.9	23.2	32.1	Per cent
18	Total ash and refuse removed (per cent of fuel used	" "	13.1	12.1	12.2	13.0	11.4	17.0	16.0	13.9	11.1	14.9	15.6	11.5	13.0	10.6	9.6	14.8	" "
	DRAFT																		
19	Ash pit	In. water	0.020	0.037	0.003	0.013	0.004	0.031	0.023	0.007	0.014	0.008	0.015	0.015	0.007	0.005	0.007	0.005	In. water
20	Stack	" "	0.012	0.040	0.016	0.046	0.068	0.022	0.030	0.021	0.039	0.063	0.015	0.024	0.028	0.030	0.045	0.008	" "
	FEED WATER																		
21	Total to boiler	Lb.	1,342	1,386	1,384	1,262	1,687	1,230	1,476	1,504	1,358	1,238	779	911	957	937	922	835	Lb.
22	Temperature	Deg. F.	68.6	67.7	63.0	64.7	76.8	68.6	65.9	64.3	65.5	68.4	75.2	73.5	71.6	71.1	73.7	73.1	Deg. F.
23	Evaporation per lb. fuel used	Lb.	6.3	6.4	6.9	6.5	6.4	6.1	7.0	7.2	6.3	6.0	3.9	4.8	4.6	4.7	4.7	4.2	Lb.
	FLUE GASES AND AIR																		
24	Carbon dioxide	Per cent	9.2	8.6	15.6	1517	16.8	12.3	13.8	15.6	15.1	14.7	8.8	10.4	11.7	13.2	12.7	11.3	Per cent
25	Oxygen	" "	11.2	11.9	4.4	4.5	2.8	7.2	5.6	3.7	4.0	4.7	10.3	8.4	6.7	4.7	4.8	7.1	" "
26	Carbon monoxide	" "	0.3	0.2	0.7	0.5	1.3	0.8	0.7	0.8	1.2	0.7	0.7	0.7	1.5	1.9	2.3	1.2	" "
27	Nitrogen, etc., by difference	" "	79.3	79.3	79.3	79.3	79.1	79.7	79.9	79.9	79.7	79.9	80.2	80.5	80.1	80.2	80.2	80.4	" "
28	Excess air content	" "	113.0	130.0	26.0	27.0	15.0	81.0	36.0	21.0	23.0	28.0	93.0	65.0	46.0	28.0	29.0	50.0	" "
29	Temperature of flue gases leaving boiler	Deg. F.	508	603	665	1034	1153	513	686	983	1030	1118	390	577	703	869	914	618	Deg. F.
30	Temperature of air entering ash pit	" "	74.9	78.7	84.9	76.2	79.7	76.1	74.9	76.6	83.6	87.2	80.2	78.5	77.9	79.7	81.3	81.1	" "
	RATES																		
31	Heat transferred to water per hr.	1000 Btu.	16	24	49	64	87	20	38	56	68	77	15	24	32	46	50	21	1000 Btu.
32	Heat transferred to water per sq. ft. of heating surface per hr.	B.t.u.	2,110	3,160	6,450	8,420	11,450	2,630	4,610	7,370	8,290	10,130	1,710	3,160	4,210	6,050	6,580	2,760	B.t.u.
33	Fuel used per hour	Lb.	2.4	3.3	6.5	8.8	12.4	3.0	4.8	7.0	9.0	11.4	2.9	4.8	6.2	8.9	9.7	4.5	Lb.
34	Fuel used per hour per sq. ft. grate area	"	1.1	1.5	2.9	4.0	5.6	1.4	2.0	3.3	4.1	5.2	1.3	2.1	2.9	4.0	4.4	2.0	"
	HEAT IN FUEL (per cent of heat in fuel charged)																		
35	Increase in heat content of fuel bed	Per cent	-4.1	-0.9	-2.4	3.7	-3.9	-3.6	-11.5	-6.9	-14.1	-1.9	-0.5	5.4	-1.3	-1.5	-1.1	2.7	Per cent
36	Total heat available(100-item 35)	" "	104.1	100.9	102.4	96.3	103.9	103.6	111.5	106.9	114.1	101.9	100.5	94.6	101.3	101.5	101.1	97.3	" "
37	Heat in combustible in ash	" "	3.1	4.7	2.2	2.8	1.5	6.3	4.1	6.3	4.4	4.7	7.8	2.7	2.9	2.0	2.4	5.1	" "
38	Heat in fuel gasified (Item 36 - item 37)	" "	101.0	104.2	100.2	93.5	102.4	98.3	107.4	102.6	109.7	97.2	92.7	91.9	98.4	99.5	98.7	92.2	" "
	HEAT BALANCE																		
39	Heat transferred to water (thermal efficiency)	Per cent	57.2	56.6	60.4	58.9	54.6	55.0	55.3	62.0	53.3	53.6	35.6	41.4	40.7	40.2	39.6	38.2	Per cent
40	Heat carried away by steam in flue gases	" "	1.0	1.0	1.0	1.3	1.2	2.5	2.4	2.6	2.5	2.9	4.7	5.0	4.9	5.2	5.4	5.1	" "
41	Heat carried away by dry flue gases	" "	20.4	25.6	19.7	25.0	24.5	12.7	16.1	19.8	22.4	25.9	10.9	15.1	15.7	18.9	19.8	14.6	" "
42	Loss due to carbon monoxide	" "	2.1	1.8	2.9	2.1	4.9	3.6	3.1	3.0	4.7	2.9	4.1	5.5	6.4	7.0	8.5	8.3	" "
43	Radiation and convection loss	" "	17.3	15.0	12.7	11.7	10.7	15.4	13.6	-12.6	10.6	10.4	12.7	10.6	9.7	8.4	8.3	10.5	" "
44	Undetermined losses	" "	2.0	0.3	3.3	1.0	1.8	9.8	5.5	0.0	6.5	4.5	32.0	24.5	22.6	20.1	18.4	24.3	" "
45	Total items 39 to 44 and heat in fuel gasified	" "	100.0	100.0	100.0	100.0	100.0	100.0	100.0	100.0	100.0	100.0	100.0	100.0	100.0	100.0	100.0	100.0	" "
	THERMAL EFFICIENCIES																		
46	Grate	Per cent	98.2	95.7	97.9	97.1	98.5	91.9	98.2	94.2	96.1	95.4	98.1	97.1	97.1	98.0	97.6	95.4	Per cent
47	Boiler and furnace (item 39)	" "	57.2	56.6	60.4	58.9	54.6	55.8	59.3	62.0	55.3	53.6	35.6	41.4	40.7	40.2	39.6	38.2	" "
48	Boiler, furnace and grate (item 46 x item 47) ÷ 100	" "	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	" "

taken by means of a mercury thermometer. It was fed to the boiler intermittently, and the level of the water in the boiler was read every half-hour. It was only by feeding the water in fairly large quantities just before reading the water-level in the gauge-glass that the priming subsided

sufficiently to enable the correct level of the water in the gauge-glass to be read. Closing the steam line momentarily also aided in reducing the fluctuation in the gauge-glass. The steam passed from the boiler to the separator, whence the water passed back to the boiler and the steam directly to the atmosphere. In computing the results, the steam was assumed to be dry. The pressure in the boiler was practically that of the atmosphere throughout. The steam temperature was measured by a mercury thermometer.

The radiation and convection loss from the outside surface of the boiler and steam separator was determined by maintaining the steam boiler at 212 deg. fahr. by means of an electric boiler attached to the main boiler through the drain-valve thus using the main boiler as a radiator. The steam outlet on the main boiler was closed. The electric current was adjusted so that the steam temperature of the main boiler remained constant at 212 deg. fahr., and therefore the input of electrical energy was dissipated by the outer surfaces of both boilers. By experiment, the radiation and convection loss of the electric boiler was found, so that the actual amount of energy dissipated at the outer surfaces of the main boiler might be separated from the total electrical energy input. The total energy input was measured by means of an ammeter and voltmeter, and for the boiler as set-up for the first series of tests the heat dissipated from the domestic boiler and separator only amounted to 2356 B.t.u. per hr.; while when the dome was added for the second series of tests, the heat dissipated amounted to 3788 B.t.u. per hr. But these losses, represent the radiation and convection losses only at zero load, and it became necessary to estimate them for the higher ratings. This was done by plotting the total unaccounted-for losses in B.t.u. per hour against the rate of evaporation for coke, anthracite, and natural gas, for which fuels the losses due to undetermined combustible gases in the flue-gases should be small, and then drawing a line from the loss at zero load which seemed to represent the probable radiation and convection loss. The loss was so determined as to give the minimum radiation and convection loss, and the same loss for the same rating was used for all fuels, which undoubtedly involved some error.

The only results computed in a somewhat unusual manner are those already referred to; the remaining results were computed in the usual manner and need not be explained here.

All observations were made every 30 min.

RESULTS OF TRIALS

TRIALS WITH FIRE-POT ONLY

The results of the trials are shown by tables and diagrams. Table 2 shows the results of the first series of trials with coal and coke, when the fire-door was sealed up and no air admitted over the fire-bed.

The heat accounts of these trials are shown in Figs. 5, 6, and 7. These heat accounts are all based on the fuel actually converted into gas to avoid any confusion that might arise through introducing the losses caused by

combustible being removed with the refuse from the ash-pit. There are thus only five curves in each figure, which divide the chart into six regions, the lowest of which represents the thermal efficiency, and those in succession above it represent the heat content of the dry flue-gas, the heat content of the superheated steam in the flue-gases, the heat lost by not burning carbon monoxide, the radiation and convection losses, and the heat unaccounted for. The thermal efficiency with anthracite and coke is seen to be fairly high for so elementary a boiler, and it does not vary very greatly with the rating, because, while the heat contents of the flue-gases increase with the load on the boiler, the radiation and convection losses decrease. But the efficiency curve for anthracite falls at the highest rating more than with coke, and this is seen to be caused by a greater variation in the heat content of the dry flue-gases with anthracite between the low and high ratings.

The heat loss caused by carbon monoxide is seen to be comparatively low, and no higher than that obtained ordinarily with a completely assembled boiler with means for admitting air over the fuel-bed.

Fig. 7 shows the heat account with bituminous coal, and it will be observed that the thermal efficiency is considerably lower than that with anthracite and coke. The principal reason for the lower efficiency is mainly the large unaccounted-for loss, which represents largely the calorific value of the gaseous hydrocarbons, hydrogen, tar, and soot which was not determined.

CHANGE OF CONTENTS OF FIRE-POT

Item 35 in Table 2 shows the increase in the heat content of the fuel-bed, and is of general importance because it shows the error which may occur in ordinary continuous testing when no means are provided for weighing the contents of the fire-pot. It shows that the greatest correction to be applied occurred in Trial 8 with anthracite, when the fire-pot contents were found to weigh less by 16 lb. (See Item 10) at the end of the trial than at the beginning, which meant that the heat given up during the trial from the stored-up energy in the fire-pot amounted to 14 per cent of the heat of the fuel charged. Had no facilities been provided for estimating this depletion of energy of the fuel-bed, the efficiencies calculated for this trial would have been about 7 per cent too high. The corrections to be applied in general, however, may be seen to be much less than this, and are less for bituminous coal than for coke and anthracite. If trials with coke and anthracite are to be run with the continuous method of stopping and starting, and no means are provided for weighing the contents of the fire-pot, they should, if accurate results are desired, be run for at least twice the length of time that these tests were run.

TRIALS WITH FIRE-POT AND DOME

Table 3 shows the complete results of the trials with the dome added to the fire-pot, when the coal was fired through the fire-door in the ordinary way. Six trials were run with coke, six with anthracite, and six

TABLE 3. RESULTS OF SECOND SERIES OF TRIALS,

Item No.	Particulars of observations	Units	Coke						Anthracite					
	Position of fire door slide		Closed	Open	Closed	Open	Closed	Open	Closed	Open	Closed	Open	Closed	Open
1	Number of trial		39	40	37	41	42	43	44	46	45	47	48	49
2	Duration of trial	Hours	74	79	29	32	21	22	69	69.5	31	32	21.5	21.5
	FUEL													
3	Calorific value per lb. as fired	B.t.u.	12,264	12,025	12,321	12,190	11,732	12,256	12,688	12,457	12,630	11,897	12,641	12,908
	Proximate analysis as fired													
4	Moisture	Per cent	0.7	0.9	0.8	1.2	1.0	1.3	1.9	3.4	3.8	3.3	3.2	3.1
5	Volatile matter	" "	2.3	1.9	2.0	2.3	3.1	2.4	5.8	5.4	5.1	4.1	5.1	7.2
6	Fixed carbon	" "	83.8	83.3	84.8	82.9	80.2	83.0	79.2	78.3	79.6	76.5	79.9	79.2
7	Ash	" "	13.2	13.9	12.4	13.6	15.7	13.3	13.1	12.9	11.5	16.1	11.8	10.5
8	Ratio fixed carbon : volatile matter		36.1	44.1	42.6	36.4	25.9	34.1	13.7	14.4	15.6	18.6	15.6	11.0
9	Total charged during trial	lb.	200	200	200	200	200	200	200	200	200	200	200	200
	FUEL BED													
	Mass of fire-pot contents													
10	Increase of mass	lb.	-1.0	1.0	4.0	-1.0	-4.0	0.0	0.0	1.0	-19.0	-2.0	-4.0	-3.0
11	Increase of carbon contents of fire-pot	"	3.0	-4.3	0.6	-6.8	-12.9	-1.6	-4.3	-6.6	-18.4	-16.9	-7.5	-3.0
12	Increase of ash contents of fire-pot	"	-4.0	5.3	4.6	5.8	8.9	1.6	4.3	7.6	-0.6	14.9	3.5	0.0
	Depth of fuel bed													
13	At start of trial	Inches	8	7	5	7	8	8	6	6	7	7	7	7
14	At finish of trial	"	7	7	6	7	8	8	5	6	6	6	7	7
15	Mean depth, before firing	"	7.7	8.0	6.4	7.3	7.9	7.3	7.1	6.9	7.9	7.3	8.1	7.4
16	Mean depth, after firing	"	15.6	15.1	14.4	15.0	14.7	15.1	12.2	10.9	13.4	11.9	13.4	11.9
	ASH AND REFUSE													
17	Carbonaceous matter in ash and refuse removed	Per cent	17.8	40.7	27.1	24.7	27.8	21.5	27.8	41.7	23.6	33.2	22.1	27.2
18	Total ash and refuse removed, per cent of fuel used	" "	18.4	19.1	14.2	14.2	15.2	16.0	15.3	15.6	14.2	12.9	12.7	14.3
	DRAFT													
19	Ash pit	In. water	.020	.017	.004	.000	.000	.000	.011	.010	.002	.000	.001	.001
20	Stack	" "	.012	.012	.015	.018	.044	.037	.004	.010	.012	.015	.049	.068
	FEED WATER													
21	Total to boiler	lb.	1227	1121	1338	1314	1314	1211	1360	1151	1500	1342	1878	1264
22	Temperature	Deg. F.	70.1	70.8	72.9	68.7	66.3	66.3	67.9	66.1	65.6	61.7	59.6	59.8
23	Evaporation per lb. fuel used	lb.	6.1	5.6	6.8	6.5	6.4	6.1	6.8	5.7	6.8	6.6	6.8	6.2
	FLUE GASES AND AIR													
24	Carbon dioxide	Per cent	7.2	4.6	11.4	8.4	11.3	9.0	10.4	5.3	14.0	9.7	14.4	9.3
25	Oxygen	" "	12.3	16.1	9.0	12.0	8.9	11.5	9.1	15.0	5.5	10.3	5.2	10.9
26	Carbon monoxide	" "	0.2	0.2	0.3	0.6	0.5	0.5	0.9	0.4	0.7	0.3	0.6	0.3
26a	Methane	" "	---	---	---	---	---	---	---	---	---	---	---	---
26b	Hydrogen	" "	---	---	---	---	---	---	---	---	---	---	---	---
27	Nitrogen (by difference)	" "	79.3	79.1	79.3	79.0	79.3	79.0	79.6	79.3	79.8	79.7	79.8	79.5
28	Excess air content	" "	171	327	75	133	73	121	75	247	35	95	32	107
29	Temperature of flue gases	Deg. F.	309	256	561	618	637	733	436	387	884	738	1076	912
30	Temperature of air entering ash pit	" "	73	76	73	75	73	73	71	70	69	75	75	81
	RATES													
31	Heat transferred to water per hour	1000 Btu	18	16	51	46	70	61	22	18	54	47	72	66
32	Heat transferred to water per sq.ft. of heating surface per hour	B.t.u.	1820	1620	5150	4650	7070	6160	2220	1820	5450	4750	7270	6670
33	Fuel used per hour	lb.	2.7	2.5	6.9	6.3	9.8	9.1	2.9	2.9	6.5	6.3	9.3	9.3
34	Fuel used per hr. per sq.ft. grate area	"	1.2	1.1	3.1	2.9	4.4	4.1	1.3	1.3	3.2	2.9	4.3	4.3
	HEAT IN FUEL (per cent of heat in fuel charged)													
35	Increase in heat content of fuel bed	Per cent	1.8	-2.6	-0.4	-4.1	-8.0	-0.9	-2.4	-3.9	-10.6	-10.3	-4.3	-1.7
36	Total heat available	" "	98.2	102.6	100.4	104.1	108.0	100.9	102.4	103.9	110.6	110.3	104.3	101.7
37	Heat in combustible in ash	" "	3.9	9.4	4.4	4.2	5.3	4.1	4.8	7.5	4.2	5.2	3.3	4.5
38	Heat in fuel gasified (Item 36 - Item 37)	" "	94.3	93.2	96.0	99.9	102.7	96.8	97.6	96.4	106.4	105.1	101.0	97.2
	HEAT BALANCE													
39	Heat transferred to water (Thermal efficiency)	Per cent	59.0	55.6	62.8	60.1	60.9	57.0	60.8	52.6	62.3	60.2	60.6	56.5
40	Heat carried away by steam in flue gas	" "	0.8	0.8	0.9	0.9	1.0	1.0	2.5	2.4	2.6	2.5	2.9	2.8
41	Heat carried away by dry flue gas	" "	13.1	15.4	16.9	24.7	26.3	27.6	12.1	20.9	20.5	25.5	25.9	32.5
42	Loss due to carbon monoxide	" "	1.8	2.8	1.7	4.5	2.9	3.5	5.0	4.4	3.0	1.9	2.5	2.0
42a	Loss due to hydrogen and hydrocarbon in flue gas	" "	---	---	---	---	---	---	---	---	---	---	---	---
42b	Loss due to tar and soot	" "	---	---	---	---	---	---	---	---	---	---	---	---
43	Loss due to radiation and convection	" "	15.1	16.3	7.9	8.1	6.4	6.4	13.7	13.7	7.6	8.0	6.3	6.2
44	Undetermined losses	" "	10.2	9.1	9.8	1.7	2.5	4.5	5.9	6.0	4.0	1.9	1.8	0.0
45	Total of items 39 to 44 or heat in fuel gasified	" "	100.0	100.0	100.0	100.0	100.0	100.0	100.0	100.0	100.0	100.0	100.0	100.0
	THERMAL EFFICIENCIES													
46	Grate	Per cent	96.0	90.9	95.6	96.0	95.1	96.0	95.3	92.7	96.2	95.2	96.9	95.6
47	Boiler and furnace (Item 39)	" "	59.0	55.6	62.8	60.1	60.9	57.0	60.8	52.6	62.3	60.2	60.6	56.5
48	Boiler, furnace and grate	" "	56.6	50.5	60.0	57.7	57.9	54.6	57.9	48.8	60.0	57.3	58.7	54.1

with bituminous coal when firing by the spreading method of firing and six when firing by the coking method.

ADMISSION OF AIR OVER THE FUEL-BED

It will be observed that the two trials at the same rating, one with the slide in the fire-door open throughout the trial and the other with it closed, are placed in adjacent columns to enable the results to be compared more readily.

Opening the slide increased the quantity of excess air in the flue gases (Item 28) as expected, but did not always reduce the loss caused by

FIRE-POT AND DOME, WITH COAL AND COKE

Bituminous, Spreading						Bituminous, Coking						Units
Closed	Open	Closed	Open	Closed	Open	Closed	Open	Closed	Open	Closed	Open	
52	54	51	53	55	56	61	62	57	58	60	59	
71.5	70	35	36.5	17.5	20	68	50.5	33.5	34	19.5	21	Hours
12,686	12,919	12,895	13,129	12,770	12,840	13,270	13,020	13,130	13,190	13,240	13,080	B.t.u.
2.1	2.0	2.3	2.0	2.2	2.0	2.1	2.1	2.7	1.9	1.9	1.9	Per cent
36.4	38.0	36.8	37.4	36.9	36.2	37.9	37.7	38.2	37.3	37.6	37.5	" "
49.9	50.2	50.7	51.4	50.1	51.0	52.1	50.4	50.7	52.5	51.8	51.0	" "
11.6	9.8	10.2	9.2	10.8	10.8	7.9	9.8	8.4	8.3	8.7	9.6	" "
1.4	1.3	1.4	1.4	1.4	1.4	1.4	1.3	1.3	1.4	1.4	1.4	
200	200	200	200	200	200	200	200	200	200	200	200	Lb.
-3.0	3.0	1.0	2.0	2.0	-3.0	3.0	-5.0	-1.5	0.0	-1.0	1.0	Lb.
-5.6	1.6	-0.7	2.2	1.3	-7.7	3.0	7.4	-2.3	0.4	-0.9	-2.2	"
2.6	1.4	1.7	0.2	0.7	4.7	0.0	2.4	0.8	-0.4	-0.1	3.2	"
6	6	6	6	6	6	5	7	5	5	5	5	Inches
6	6	5	6	6	5	6	6	5	6	5	5	"
6.3	6.1	6.3	6.5	6.2	6.9	---	---	---	---	---	---	"
11.0	9.4	11.1	12.3	10.1	11.4	---	---	---	---	---	---	"
27.9	34.7	17.6	25.4	10.6	18.6	34.0	36.4	18.1	25.9	16.8	25.7	Per cent
14.0	14.8	11.3	12.6	11.9	10.0	12.2	13.2	9.7	11.5	10.4	10.8	" "
.016	.012	.004	.001	.000	.000	.014	.009	.000	.001	.000	.000	In. water
.018	.014	.026	.025	.036	.059	.009	.011	.010	.006	.043	.043	" "
964	908	1107	968	987	1053	890	1008	1011	1072	995	1034	Lb.
67.4	68.1	64.4	66.7	59.6	61.4	66.9	72.7	58.6	61.8	59.4	60.0	Deg. F.
4.7	4.6	5.6	4.9	4.7	5.2	4.8	4.9	5.0	8.4	5.0	5.2	Lb.
7.1	5.5	11.5	8.1	14.5	11.6	9.9	6.8	12.9	10.7	14.9	12.1	Per cent
12.4	14.4	7.3	11.1	2.9	7.3	9.1	12.8	4.6	8.2	3.2	6.9	" "
0.6	0.4	0.8	0.6	2.4	0.6	1.5	0.7	2.5	0.5	2.8	0.7	" "
---	---	---	---	---	---	0.4	0.1	0.6	0.1	0.3	0.2	" "
---	---	---	---	---	---	0.4	0.1	0.8	0.1	0.7	0.2	" "
79.9	79.7	80.4	80.2	80.2	80.5	78.7	79.5	78.6	80.4	78.7	79.9	" "
140	212	82	109	16	52	77	184	28	62	18	48	" "
406	447	874	649	1050	1000	486	503	777	841	1130	1101	Deg. F
74	77	72	74	81	72	74	79	71	75	78	78	" "
15	14	36	30	60	59	16	19	34	35	87	58	1000 Btu.
1520	1410	3640	3030	6060	5960	1620	1920	3430	3640	5760	5580	B.t.u.
2.9	2.9	5.7	5.6	11.4	10.0	3.2	3.4	6.0	5.9	10.3	9.5	Lb.
1.3	1.3	2.6	2.5	5.1	4.6	1.5	1.6	2.7	2.7	4.7	4.3	"
-3.2	1.0	-0.4	1.2	0.7	-4.4	1.7	-4.1	-1.3	0.2	-0.5	-1.2	Per cent
103.2	99.0	100.4	98.8	99.3	104.4	98.3	104.1	101.3	99.8	100.5	101.2	" "
4.6	5.5	2.2	3.5	1.4	1.8	4.5	5.5	1.9	3.3	1.9	3.1	" "
98.6	93.5	98.2	95.3	97.9	102.6	93.8	98.6	99.4	96.5	98.6	98.1	" "
42.9	41.9	48.9	43.1	41.9	44.8	39.7	43.6	43.5	47.2	42.8	45.2	Per cent
4.4	4.7	5.2	4.9	5.5	5.2	4.7	4.5	5.0	5.2	5.4	5.5	" "
16.1	20.6	22.1	22.1	20.4	26.0	12.7	19.7	16.5	23.3	24.7	24.8	" "
4.4	3.8	3.6	3.8	8.0	2.8	6.5	5.6	7.6	2.2	6.0	2.9	" "
---	---	---	---	---	---	6.3	3.8	6.8	2.2	3.9	1.9	" "
---	---	---	---	---	---	7.0	3.6	7.3	3.2	5.0	1.4	" "
13.0	13.1	7.7	7.6	4.8	5.2	11.5	10.8	7.2	7.8	6.0	5.4	" "
21.2	15.9	12.5	10.5	19.4	16.0	11.6	8.7	6.1	9.2	7.2	13.2	" "
100.0	100.0	100.0	100.0	100.0	100.0	100.0	100.0	100.0	100.0	100.0	100.0	" "
98.6	94.5	97.8	96.4	98.6	98.3	90.5	94.7	98.1	96.7	98.1	97.0	Per cent
42.9	41.9	48.9	43.1	41.9	44.8	39.7	43.6	43.5	47.2	42.8	45.2	" "
41.1	39.6	47.8	41.6	41.3	44.1	37.9	41.2	42.7	45.6	42.0	43.8	" "

carbon monoxide and the other combustible gases, tars, and solids leaving the fuel-bed. Thus, throughout the trials with coke it will be observed that the loss due to carbon monoxide increased when secondary air was admitted, and although the secondary air somewhat reduced this loss when burning anthracite, the heat gained by burning it was less than the heat lost by the additional air brought in to burn it.

EFFECT OF ADMITTING AIR IN SOFT COAL TRIALS

But on burning bituminous coal fired by the spreading method, at the highest rating, and in all the trials when firing it by the coking method, the secondary air so reduced the losses due to incomplete combustion as to raise the efficiency materially.

Little or no improvement in combustion, however, was caused by admitting secondary air at the lowest ratings when firing by the spreading method, so that the additional air increased the heat contents of the escaping gases and lowered the efficiency.

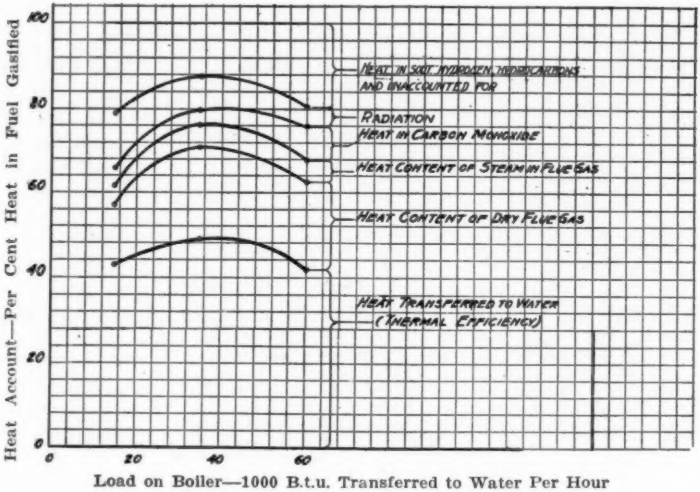


FIG. 12. HEAT ACCOUNT BASED ON HEAT IN FUEL GASIFIED. BITUMINOUS COAL FIRED BY SPREADING METHOD, SECOND SERIES, FIRE-POT AND DOME, NO SECONDARY AIR, PORTS IN FIRE-DOOR CLOSED

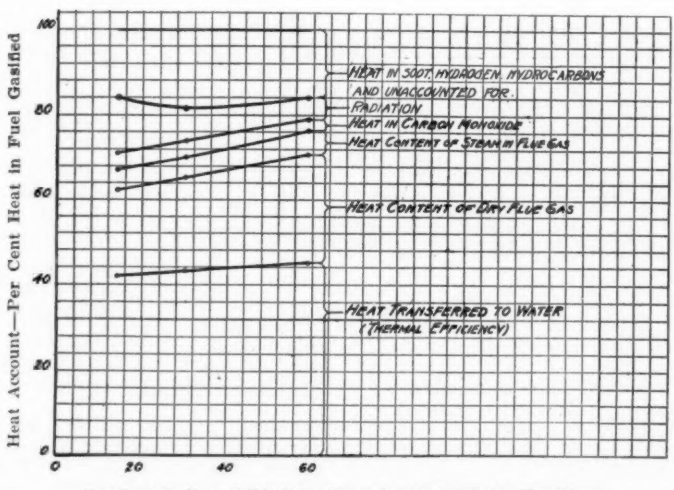


FIG. 13. HEAT ACCOUNT BASED ON HEAT IN FUEL GASIFIED. BITUMINOUS COAL FIRED BY SPREADING METHOD, SECOND SERIES, FIRE-POT AND DOME, SECONDARY AIR PORTS IN FIRE-DOOR WIDE OPEN

TABLE 4. RESULTS OF NATURAL GAS TRIALS

Item No.	Particulars of observation	Units	Fire pot only							Fire pot and dome						
1	Number of trial		28	29	27	25	26	24	23	35	33	34	32	31	30
2	Duration of trial	Hours	2.0	2.0	3.0	2.5	2.5	3.0	3.0	3.0	3.0	3.0	3.0	3.0	2.0
	FUEL														
3	Calorific value per cu.ft. at 32°F. and 14.7 lb. per sq. in.	B.t.u.	1110	1110	1110	1130	1122	1130	1130	1118	1118	1118	1118	1118	1118
	Fuel gas analysis, by volume														
4	Methane	Per cent	89.7	89.7	89.7	88.4	88.2	88.4	88.4	89.6	89.6	89.6	89.6	89.6	89.6
5	Ethane	" "	8.3	8.3	8.3	10.1	9.8	10.1	10.1	8.8	8.8	8.8	8.8	8.8	8.8
6	Nitrogen (by difference)	" "	2.0	2.0	2.0	1.5	2.0	1.5	1.5	1.6	1.6	1.6	1.6	1.6	1.6
7	Gas pressure at burner	In. water	6.5	6.5	8.5	11.6	9.6	11.3	13.5	6.9	12.0	11.8	9.8	8.9	11.0
8	Total gas burned during test (at 32°F. and 14.7 lb. per sq. in.)	Cu. ft.	40.2	41.7	128.0	167.0	177.0	252.0	323.0	61.5	127.0	126.0	223.0	266.0	209.0
	DRAFT														
9	Under burner	In. water	0.016	0.013	0.018	0.017	0.019	0.032	0.019	0.019	0.018	0.013	0.036	0.023	0.027
10	Stack	" "	0.002	0.000	0.001	0.005	0.005	0.017	0.008	0.004	0.003	0.000	0.020	0.007	0.018
	FEED WATER														
11	Total to boiler	Lb.	16.0	20.5	73.0	95.0	98.0	139.0	172.0	34.0	72.0	80.0	141.0	161.0	130.5
12	Temperature	Deg. F.	77	80	78	79	80	76	76	78	72	80	76	76	79
13	Pounds of water per cu. ft. of gas (at 32°F. and 14.7 lb. per sq. in.)	Lb.	0.40	0.49	0.57	0.57	0.56	0.55	0.53	0.55	0.57	0.64	0.63	0.60	0.62
	FLUE GASES AND AIR														
14	Carbon dioxide	Per cent	5.5	9.2	8.2	10.2	9.5	11.0	11.0	7.5	6.8	9.1	10.0	10.8	11.0
15	Oxygen	" "	10.8	4.3	6.5	2.9	4.3	1.4	1.6	7.4	8.8	4.3	3.5	1.6	1.3
16	Carbon monoxide	" "	0.2	0.1	0.0	0.1	0.0	0.0	0.1	0.0	0.0	0.3	0.0	0.1	0.2
17	Nitrogen (by difference)	" "	83.5	86.4	85.3	86.8	86.2	87.6	87.3	85.1	84.4	86.3	86.5	87.5	87.5
18	Excess air	" "	95.0	23.0	40.0	14.0	23.0	6.0	7.0	49.0	65.0	23.0	16.0	7.0	6.0
19	Temperature of flue gases leaving boiler	Deg. F.	459	565	767	975	976	1036	1205	359	618	709	923	1012	1099
20	Temperature of air entering below burner	" "	84	87	92	85	92	84	85	82	73	81	79	80	87
	RATES														
21	Heat transferred to water per hr.	1000 B.t.u.	9	11	27	42	43	51	63	13	27	29	52	59	72
22	Heat transferred to water per sq. ft. of heating surface per hr.	B.t.u.	1180	1450	3550	5530	5660	6710	8290	1310	2730	2930	5250	5960	7270
	HEAT BALANCE														
23	Heat transferred to water (Thermal efficiency)	Per cent	39.7	48.8	56.9	55.6	54.6	54.1	52.2	54.6	56.2	62.9	62.5	59.8	61.7
24	Heat carried away by steam in flue gases	" "	11.2	11.7	12.5	13.4	13.4	13.7	14.4	10.8	12.0	12.4	13.3	13.7	14.0
25	Heat carried away by dry flue gases	" "	12.4	9.8	15.6	16.6	17.8	16.6	19.4	7.0	15.1	12.7	16.1	16.3	17.3
26	Loss due to carbon monoxide	" "	1.1	3.5	0.0	0.3	0.0	0.0	0.3	0.0	0.0	1.0	0.0	0.3	0.6
27	Radiation and convection losses	" "	17.1	17.9	14.3	12.2	11.8	11.2	10.4	19.5	10.9	11.3	7.8	6.9	6.4
28	Undetermined losses	" "	18.5	8.3	0.7	1.9	2.4	4.4	3.3	8.1	5.7	-0.3	0.3	3.0	0.0
29	Total of items 23 to 28 or heat in fuel	" "	100.0	100.0	100.0	100.0	100.0	100.0	100.0	100.0	100.0	100.0	100.0	100.0	100.0

TABLE 5. COMPOSITION OF THE PRODUCTS OF COMBUSTION OF BITUMINOUS COAL FIRED BY THE SPREADING METHOD FIRE-POT ONLY

Item No.	Particulars of observation	Units	Fire-pot only. Spreading method of firing.									
1	Trial number		20	20	17	17	21	21	18	18	15	15
2	Rate, pounds of fuel fired per hour	Lb.	2	2	4	4	4	4	6	6	10	10
3	Time of sampling relative to firing		1 hr.BF	1 hr.AF	1 hr.BF	1 hr.AF	1 Hr.BF	1 hr.AF	1 hr.BF	1 hr.AF	1 hr.BF	1 hr. AF
4	Duration of sampling	Minutes	30	20	42	25	21	30	26	20	27	45
5	Stack draft (depression)	In. water	-0.006	0.005	0.015	0.015	-0.011	-0.006	0.005	-0.010	0.020	0.089
6	Flue gas temperature	Deg. F.	338	355	557	637	918	695	653	748	1045	863
7	Weight of tar per cu. ft.	Grams	0.002	0.129	0.004	0.165	0.010	0.071	0.004	0.885	0.002	0.115
8	Weight of soot per cu. ft.	"	0.014	0.033	0.023	0.049	0.012	0.030	0.008	0.677	0.003	0.091
	FLUE GASES, per cent by volume											
9	Carbon dioxide	Per cent	9.4	7.3	6.5	8.7	8.9	10.1	5.6	12.1	11.3	12.2
10	Carbon monoxide	" "	0.9	0.7	0.5	1.4	1.2	1.9	0.3	1.9	0.2	3.1
11	Methane	" "	0.0	0.2	0.0	0.6	0.2	0.4	0.0	0.7	0.0	1.5
12	Hydrogen	" "	0.3	0.5	0.2	0.9	0.7	0.6	0.1	1.2	0.1	1.1
13	Oxygen	" "	10.2	12.6	12.8	10.0	10.1	8.5	14.7	5.9	9.2	3.0
14	Nitrogen (by difference)	" "	79.2	78.7	80.0	78.4	78.9	78.5	79.3	78.2	79.2	79.1
15	Ratio, air supplied to air required		1.82	1.97	2.37	1.40	1.69	1.28	3.15	0.70	1.52	0.85
	HEAT LOSSES. Per cent heat in fuel gasified											
16	Heat in CO, CH_4, and H_2	Per cent	6.2	10.3	5.1	17.6	12.4	14.6	3.8	11.3	1.4	25.4
17	Heat in tar and soot	" "	0.9	11.0	2.1	10.9	1.2	4.7	1.3	39.2	0.3	6.9
18	Total undeveloped heat (Items 16 and 17)	" "	7.1	21.3	7.2	28.5	13.6	19.3	5.1	50.5	1.7	32.3

NOTE: BF - Before firing AF - After firing

TABLE 6. COMPOSITION OF THE PRODUCTS OF COMBUSTION OF BITUMINOUS COAL FIRED BY THE SPREADING METHOD FIRE-POT AND DOME

Item No.	Particulars of observation	Units	Fire-pot and dome Spreading method of firing.														
	Position of fire door slide		Closed	Closed	Closed	Closed	Closed	Closed	Closed	Open	Open	Open	Open	Closed	Closed	Open	Open
1	Trial number		52	52	52	51	51	51	51	53	53	53	53	55	55	56	56
2	Rate, pounds of fuel fired per hour	Lb.	2	2	2	6	6	6	6	6	6	6	6	10	10	10	10
3	Time of sampling relative to firing		AF	BF	AF	BF	AF	BF	AF	BF	AF	1 hr.AF	2 hr.AF	BF	AF	BF	AF
4	Duration of sampling	Minutes	18	18	18	19	18	19	19	20	18	20	18	18	15	13	21
5	Stack draft (depression)	In.water	0.020	0.008	0.005	0.016	0.018	0.018	0.018	0.020	0.040	0.020	0.020	0.025	0.018	0.037	0.038
6	Flue gas temperature	Deg.F.	485	528	268	748	1119	881	418	812	1031	549	472	1170	437	1212	1239
7	Weight of tar per cu. ft.	Grams	0.234	0.034	0.215	0.022	0.083	0.003	0.174	0.004	0.133	0.115	0.064	0.009	0.829	0.002	0.082
8	Weight of soot per cu. ft.	"	0.045	0.020	0.032	0.018	0.061	0.008	0.079	0.008	0.041	0.027	0.017	0.014	0.097	0.009	0.065
	FLUE GASES, per cent by volume																
9	Carbon dioxide	Per cent	9.0	10.2	6.0	8.5	12.4	11.9	7.4	7.1	8.5	7.5	4.6	16.8	8.7	13.6	12.5
10	Carbon monoxide	" "	1.5	1.0	2.7	0.7	2.6	0.0	1.4	0.3	1.4	1.9	1.1	0.5	6.5	0.1	2.0
11	Methane	" "	0.8	0.1	0.6	0.1	0.7	0.0	0.3	0.0	0.2	0.3	0.2	0.0	2.5	0.0	0.6
12	Hydrogen	" "	0.2	0.1	0.9	0.0	0.7	0.0	0.4	0.0	0.3	0.5	0.3	0.0	2.7	0.0	0.1
13	Oxygen	" "	9.7	8.9	12.8	11.3	3.5	7.6	11.7	12.8	10.2	11.7	15.0	3.0	7.1	6.8	4.8
14	Nitrogen (by difference)	" "	78.8	79.7	77.0	79.4	80.1	80.5	78.8	79.8	79.4	78.1	78.8	79.4	72.5	79.5	80.0
15	Ratio, air supplied to air required		1.32	1.57	1.63	1.98	0.97	1.54	1.66	2.41	1.56	1.73	2.75	1.14	0.66	1.47	1.07
	HEAT LOSSES. Per cent heat in fuel gasified																
16	Heat in CO, CH_4, and H_2	Per cent	16.9	6.6	26.8	5.8	17.4	0.0	13.5	2.2	11.1	16.7	16.7	16.2	38.3	0.4	13.4
17	Heat in tar and soot	" "	13.5	2.8	14.5	2.5	5.3	0.6	14.6	0.9	9.7	8.3	7.8	0.8	25.3	0.5	5.7
18	Total undeveloped heat (Items 16 and 17)	" "	30.4	9.4	41.3	8.3	22.7	0.6	28.1	3.1	20.8	25.0	24.5	17.0	63.6	0.9	19.1

NOTE: BF - Before firing AF - After firing

TABLE 7. COMPOSITION OF THE PRODUCTS OF COMBUSTION OF BITUMINOUS COAL FIRED BY THE COKING METHOD FIRE-POT AND DOME

Item No.	Particulars of observation	Units	Fire-pot and dome Coking method of firing										
	Position of fire door slide		Closed	Closed	Closed	Closed	Closed	Open	Open	Open	Open	Open	Open
1	Trial number		57	57	57	57	57	58	58	58	58	58	58
2	Rate, pounds of fuel per hour	Lb.	6	6	6	6	6	6	6	6	6	6	5
3	Time of sampling relative to firing		AF	1 hr.AF	2 hr.AF	3 hr.AF	4 hr.AF	AF	1 hr.AF	2 hr.AF	3 hr.AF	4 hr.AF	5 hr.AF
4	Duration of sampling	Minutes	30	30	30	30	30	25	28	18	29	26	25
5	Stack draft (depression)	In. water	-0.010	-0.005	0.000	0.001	0.002	0.022	0.006	0.006	0.008	0.005	0.017
6	Flue gas temperature	Deg. F.	570	464	525	712	881	668	933	968	744	747	455
7	Weight of tar per cu. ft.	Grams	0.376	0.156	0.093	0.125	0.081	0.019	0.008	0.004	0.026	0.087	0.009
8	Weight of soot per cu. ft.	"	0.106	0.043	0.037	0.049	0.118	0.013	0.024	0.048	0.015	0.108	0.007
	FLUE GASES, per cent by volume												
9	Carbon dioxide	Per cent	11.3	8.5	8.9	11.9	15.9	8.2	14.4	11.2	8.9	12.1	4.2
10	Carbon monoxide	" "	2.8	1.4	0.6	1.9	3.0	0.5	0.4	0.6	0.6	0.6	0.4
11	Methane	" "	1.2	0.6	0.4	0.6	0.2	0.0	0.1	0.0	0.0	0.0	0.2
12	Hydrogen	" "	1.5	0.7	0.5	0.9	0.0	0.0	0.0	0.0	0.3	0.2	0.0
13	Oxygen	" "	5.3	9.8	10.7	6.2	2.5	10.4	4.1	7.4	10.4	5.3	15.7
14	Nitrogen (by difference)	" "	77.9	79.0	78.9	78.5	78.4	80.9	81.0	80.8	79.8	81.8	79.5
15	Ratio, air supplied to air required		0.87	1.40	1.66	1.12	0.96	1.85	1.19	1.46	1.83	1.19	3.46
	HEAT LOSSES. Per cent of heat in fuel gasified												
16	Heat in CO, CH_4, and H_2	Per cent	22.8	17.6	6.7	15.8	9.8	3.1	2.5	2.8	5.0	3.1	11.0
17	Heat in tar and soot	" "	16.5	10.6	7.5	7.0	5.8	2.2	1.3	2.4	2.6	8.4	2.0
18	Total undeveloped heat (Items 16 and 17)	" "	39.3	28.2	14.2	22.8	15.6	5.3	3.8	5.2	7.6	11.5	13.0

NOTE: After firing — AF

Evidently leaving the slide open continuously when burning anthracite and coke is not to be recommended for this size of boiler since more than sufficient air passes around the walls of the fire-pot to burn the combustible gases arising from the fuel-bed, and any additional air may actually retard combustion. But since the loss due to carbon monoxide with anthracite and coke amounts to about 2 to 5 per cent of the heat in the fuel gasified, there is an obvious field for economy in the design of some device that will mix the combustible gases and air more thoroughly in some region near the surface of the fuel-bed where they may be ignited.

The reduction in the combustible losses with increased air supply at the highest rating when firing bituminous coal by the spreading method is attributed to the higher temperatures and velocities at the higher ratings which permit the gases to burn more readily, and a similar reduction when firing by the coking method is attributed to the hot surfaces of the exposed incandescent fuel on one side of the fire-pot aiding in igniting the combustible gases.

Figs. 8, 9, 10, and 11 show graphically the heat accounts based on the fuel gasified for coke and anthracite when using the fire-pot and dome; those with the air-ports closed are shown on charts sepa-

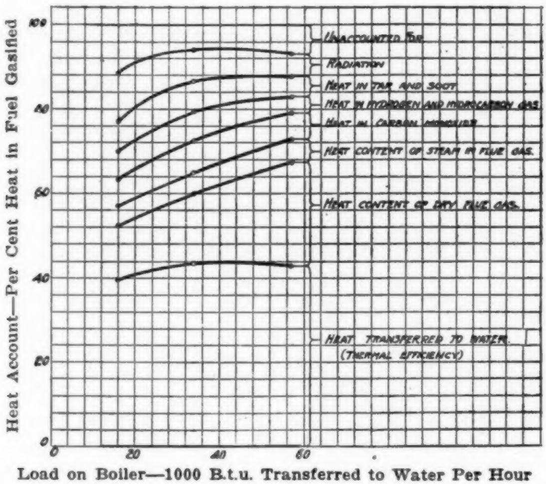


FIG. 14. HEAT ACCOUNT BASED ON HEAT IN FUEL GASIFIED. BITUMINOUS COAL, FIRED BY COKING METHOD, SECOND SERIES, FIRE-POT AND DOME, NO SECONDARY AIR, PORTS IN FIRE-DOOR CLOSED

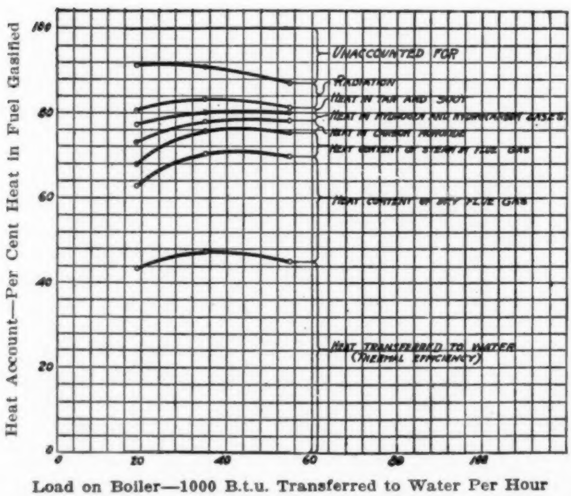


FIG. 15. HEAT ACCOUNT BASED ON HEAT IN FUEL GASIFIED. BITUMINOUS COAL, FIRED BY COKING METHOD, SECOND SERIES, FIRE-POT AND DOME, SECONDARY AIR PORTS IN FIRE-DOOR WIDE OPEN

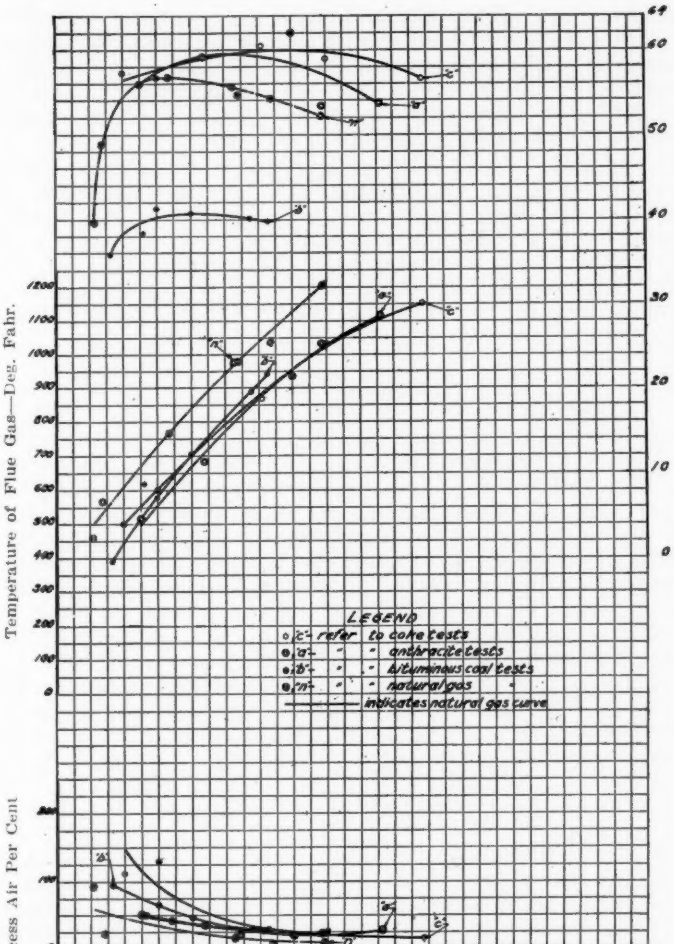


FIG. 16. CURVES SHOWING THE THERMAL EFFICIENCIES BASED ON FUEL GASIFIED, THE TEMPERATURES AND EXCESS AIR CONTENTS OF THE FLUE GASES, FOR THE VARIOUS FUELS WHEN USING THE FIRE-POT ONLY

rate from where they are open. With the air-ports closed it will be seen that the efficiency is somewhat higher than when open, that it varies little with the rating, and has a mean value a little over 60 per cent.

Figs. 12 and 13 show graphically the heat accounts for the bituminous coal tests with and without the admission of secondary air when firing

by the spreading method. Here the efficiency is seen to be much lower than when firing anthracite and coke, principally because the unaccounted-for losses are much higher and in a less degree on account of the greater loss due to the greater quantity of steam formed from the bituminous coal. The higher unaccounted-for losses are largely due to the presence of gaseous hydrocarbons, hydrogen, tar, and soot in the flue-gases.

Figs. 14 and 15 show the heat-account charts for the bituminous coal tests when fired by the coking method. They are particularly interesting because they show the losses due to hydrogen, gaseous hydrocarbons, tar, and soot, and by including these losses the undetermined loss is considerably reduced, though it still remains much larger than desirable. Probably the high undetermined loss is partly caused by using too low a value for the radiation and convection loss, though it is impossible for this to account for more than a small portion of the undetermined losses.

NATURAL GAS TRIALS

Table 4 shows the results of the trials with natural gas. These tests were carried out primarily to obtain values for the radiation and convection losses when the boiler was operated at a fairly high rating. The high unaccounted-for losses at the lower rates of steaming are probably caused by an unsteady flame, which permitted some gas to pass off unburned and undetected by analysis. In operating this boiler, great care was required in adjusting the air supply in order to maintain the excess air at a value to ensure a fairly good efficiency. Had the boiler been operated without carefully adjusting the secondary air supply, the efficiency would have been very low; and since with careful adjustment the over-all efficiency was only about that obtained with anthracite and coke, the necessity of having a carefully designed burner, whereby the air supply is automatically and correctly adjusted, is obvious if a reasonable efficiency is to be maintained.

COMPARISON OF FUELS

Fig. 16 shows the thermal efficiencies based on the fuel gasified, the temperatures and excess-air contents of the flue-gases for the various fuels at various loads when using the fire-pot only, with no secondary air. It will be observed that the efficiencies with coke, anthracite, and natural gas are considerably higher than those with bituminous coal. The flue-gas temperatures with natural gas are higher than those of the solid fuels, probably because the natural-gas flame radiated less of its energy than the solid fuels. The curves showing the excess air all show how it decreased with the load on the boiler.

Fig. 17 shows similar curves for the trials with the fire-pot and dome. The best efficiencies were obtained with anthracite and coke when no secondary air was admitted through the fire-door. The mean thermal efficiency with bituminous coal when fired by the spreading method is shown to be higher when no air was admitted over the fuel-bed. To avoid confusion the corresponding curves for the trials when fired by

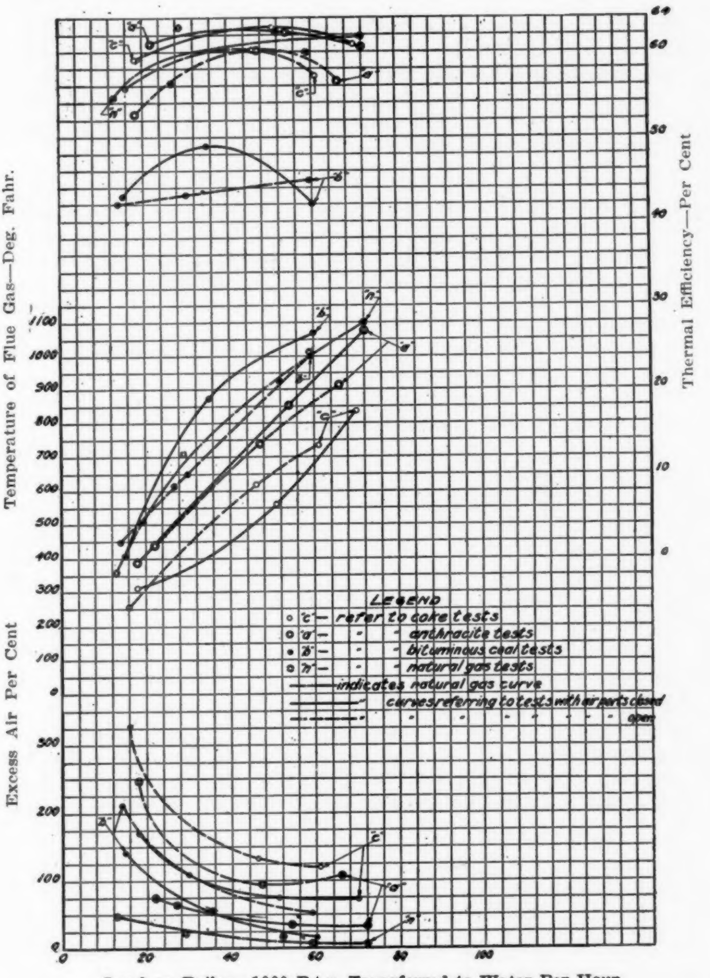


FIG. 17. CURVES SHOWING THE THERMAL EFFICIENCIES BASED ON FUEL GASIFIED, THE TEMPERATURES AND EXCESS AIR CONTENTS OF THE FLUE GASES, FOR THE VARIOUS FUELS WHEN USING THE FIRE-POT AND DOME

the coking method are not shown. When firing by the coking method the efficiency was higher when air was admitted over the fuel-bed. The flue-gas temperatures with bituminous coal are shown to be higher than those with anthracite, which in turn are higher than those with coke.

The marked reduction of excess air with the load is again shown by this chart.

So far, the results of the trials as a whole have been considered, but it is important, particularly when firing bituminous coal, to know how the combustion changes between the intervals of charging the fuel. With this end in view, tables have been prepared showing the composition of samples of flue gas from time to time, and the variation in the heat lost by not burning its combustible constituents.

Table 5 shows the composition of the gases for the first series of trials with bituminous coal, when no air was admitted over the fuel-bed. It will be observed that the samples were taken in pairs and that each pair represents one sample taken before firing and another taken after firing. Generally, the total heat remaining latent as combustible gases, vapors, and solids (total undeveloped heat, Item 18), is considerably greater after than before firing; but Item 15 shows that at all times when burning the coal at the rates of 2 to 4 lb. per hr. ample air was supplied to burn it. At the two higher rates, however, insufficient free oxygen was present after firing to burn the combustible constituents, even supposing that all of the oxygen supplied could be used up.

Table 6 shows a similar set of gas analyses for some of the trials on the fire-pot and dome when firing by the spreading method. Here again will be found wide variations in the quantities of combustible constituents of the gases, before and after firing. No very noticeable difference in the undeveloped heat between the trials with and without secondary air is apparent, except at the highest rating (Trials 55 and 56), when, after firing with no secondary air, 64 per cent of the heat of the fuel gasified went up the stack as combustible matter, and in the corresponding trial when secondary air was admitted, less than 20 per cent of this heat escaped.

In Table 7, the composition of the gas is given for two periods when firing 6 lb. of coal per hour by the coking method. The first set of samples were taken during Trial 57, when the ports in the fire-door were closed, and the second set during Trial 58 when they were left open. The loss caused by the combustible constituents is considerably greater when no air was admitted through the air-ports.

CONCLUSIONS

It is interesting to note that when firing by the coking method, with the slots in the fire-door open, the losses caused by combustible gradually increased after the first hour, whereas, with no secondary air, they tended to decrease.

The wide variation in the quality of the gases throughout any period of firing, as shown by these tables, shows the impossibility of automatically providing for a supply of secondary air over the fuel-bed which will give the highest boiler efficiency; but if means could be provided for the better use of the free oxygen already present in haphazard, though as a rule excessive, quantities, the efficiency would be considerably increased.

DISCUSSION

H. M. NOBIS: Would the boiler be improved if projections were cast on the inside?

W. M. MYLER: As the different types of fuel burn differently the manufacturer of this boiler has made us some special sections with the idea of increasing the combustion space above the fuel bed, rather than with projections as suggested by the previous speaker. It is thought that if it is possible to get combustion to take place before the flame strikes the water-cooled surface, the thermal efficiency can probably be raised. We have five of these special intermediate sections to be tried out.

P. J. DOUGHERTY: One thought in this vaper I would like to stress in the interest of designing and operating engineers is in reference to fuel economy and the excess air over the fuel bed. Mr. Myler has brought out very forcibly that the percentage of excess air over the fuel bed was very high in order to produce economical combustion conditions. He has brought out the fact that the heat loss by not burning carbon monoxide was not more than 5 per cent and also that the excess air over the fire reduced the carbon dioxide content, which of course produces a very heavy loss. We know that in burning soft coal that excess air over the fire should never as a rule continue more than two to five minutes or holes will be burned through the fuel bed which according to results of a survey by Mr. Kreisinger demonstrated quite conclusively that in our best modern power plants 40 per cent of the fuel waste is due to excess air rather than lack of air.

JOHN HOWATT: A furnace has to be designed for the kind of fuel to be burned, though the paper does not say anything about the size of fuel to be used. In the case of coke those who sell it recommend the use of the small pea size with it to prevent excess air. Test heating plants designed to care for a heating load on a zero day when used under average temperature conditions of 36 deg. results in a big waste of fuel during the spring and fall months.

J. M. ROBB: The tests show a great variation of efficiency under a slight variation in draft and if this is so any method used to maintain a constant intensity of draft in the chimney would tend to bring more uniform operating results.

W. M. MYLER: Because the stack used in our gas tests was intended for testing considerably larger boilers it was necessary to put in an auxiliary stack to run low rates. Since the preparation of this paper we have run a series of gas tests and found that we got better results with pressures ranging from one to two thousandths of an inch than was possible with any draft.

E. S. HALLETT: Anything that would save a hundred tons of coal is important to us in St. Louis. Some time back we had 130 portable schools with separate furnaces and we encountered the same difficulty, mentioned by Mr. Dougherty, of experiencing the greatest fuel loss when there was

a low temperature above the fuel bed. We solved this problem very satisfactorily by consolidating eight or ten portable schools into a group and heating a group by means of one furnace. The advantages, from the standpoint of fuel economy, labor economy, and general satisfaction is such that we would never think of going back to a single heater in each room.

No. 664

HEATING WITH CONDENSER CIRCULATING WATER

By John E. Williams[1], Waterbury, Conn.

Non-Member

THE heating of a plant like the Chase Metal Works is a problem, for the amount of glass used in its roof of modern saw-tooth construction and its huge rooms, make the heating a serious question for consideration by the modern engineer. It has been solved by installing a heating system which at the time it was constructed was the only one of its kind in the country. It is done by connecting over 78 miles of pipe directly onto the condensers in the power house, by pumping hot water through the mill, returning it to the condenser again, reheating it and recirculating it through the mill.

In all, over 36 acres of buildings are heated with a partial vacuum on the turbines and engines which receive all the B.t.u. sent to them either in work or heat, maintaining a full load irrespective of the heat requirements, due to various temperatures in the hot-water system and part of this being done by substituting surface condensers for heaters.

Heating under partial vacuum has not received the attention that it deserves. Twenty-five years ago at the Chase Rolling Mills a heater was set up between a 48 in. low-pressure cylinder and the jet condenser. A 50,000 sq. ft. open forced-feed heating system is connected to this heater.

In the fall it is necessary to curtail injection water and to run the condenser at about 23 in. of vacuum. When heat is called for, the circulating pump is started up forcing through this heater 50,000 gal. of water per hr. The exhaust is then passed through this heater and the vacuum goes up to 26 in. The buildings are also being heated with the heat due to 26 in. of vacuum.

As the weather gets colder, the water is cut down on the jet condenser. This plan is continued, meeting the weather requirements until the vacuum is reduced to 20 in. Now there is but a small amount of injection water in the condenser, for the heater has been practically made into

[1] Chief Engineer, Chase Metal Works, Waterbury, Conn.

Paper presented at the Semi-Annual Meeting of American Society of Heating and Ventilating Engineers, Chicago, Ill., May, 1923.

a surface condenser and the jet condenser has become a wet vacuum pump.

The engine is now cutting-off later to provide for the loss in vacuum but it should be borne in mind that all of the heat is being gotten back minus the work expanded and the engine radiation, as the condensate water is used for boiler feed or manufacturing purposes.

The same heating water, of course, continues to circulate through this system, picking up its heat as it goes through this heater and radiating it as it goes through the heat coils in the plant. The same water after it leaves the heater goes through an economizer and is here stepped up. If the assumption is made that the heating medium is at 20 in. of vacuum and 158 deg. fahr., when it leaves the economizer it is at a temperature of 166 deg. fahr., which is sufficient to meet an outdoor temperature with the radiation ratio of 60 to 1 of 16 deg. fahr. above zero. As the weather in this locality is below this temperature but a short time of the season,

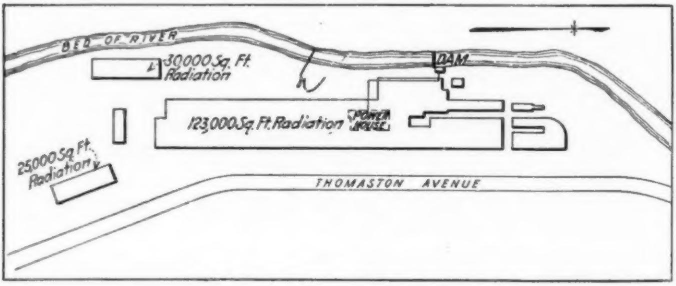


FIG. 1. SKETCH SHOWING LOCATION OF HEATED BUILDINGS AT CHASE METAL WORKS

it is found that with this system, that rather than to lower the vacuum below the 20 in. to supply the extra heat required for this lower outdoor temperature (as this heating water is continually passing through a live steam heater which also is an exhaust heater), this extra heat can be supplied by turning in the exhaust from a steam compressor or a small steam engine.

At the same mill there is another system working on 110,000 sq. ft. of radiation in about the same manner except that it is run from 6 in. of vacuum to 26 in. covering the full range of heat until partial vacuum. This 110,000 foot system was started with a 16 in. high-pressure engine exhausting into one of these heaters under about 1 lb. back pressure. The condensate was pumped back into the boilers. The heat could be regulated only by cutting some of the load off, or putting more on, but the flexibility of the partial vacuum was not there, neither was the power.

It is often remarked that any old engine is satisfactory as long as the heat in the exhaust is gotten back. This is true, but this is not of as much interest as the amount of power to be obtained from this exhaust. It is not our claim that we are getting our heat for nothing, but we do

claim that we get our power from these heating engines and turbines for about 10 per cent of the steam that is sent through them and can use the 90 per cent for heating.

A 500 hp. cross-compound engine with the same size high-pressure cylinder, namely 16 in., calling for the same amount of steam, took the place of this high-pressure system. The compound was connected up to run on the partial vacuum principle, and instead of 35,000 k.w. hr. per week, we get from 60,000 to 70,000 k.w. hr. per week, which makes quite a difference in the economy of the plant.

Twelve years ago when erecting the Chase Metal Works with its modern saw-tooth and glass buildings, the heating with turbines was found to be quite a different problem from heating with the steam engines.

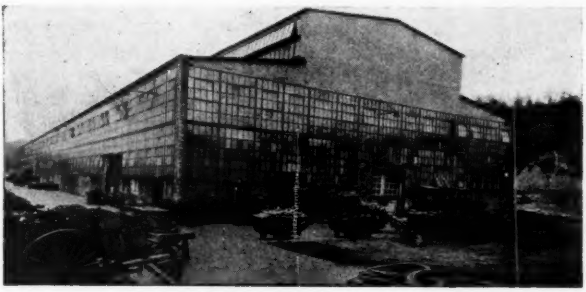


FIG. 2. BUILDING 100 x 400 FT. HEATED BY 30,000 SQ. FT. OF RADIATION

Although it is practical to connect to the steam engines 18 in. pipes and valves for a separate heating system, we could not see it in this light in connection with the cumbersome exhaust pipes and the room this separate heating system under partial vacuum would take up in connection with steam turbines. So, we got this system by doing away with it.

We would not, of course, care to install a heating system with one heater and one circulating pump, as the requirement is for one exhaust heater and one live-steam heater with two circulating pumps. So we put in at this mill the same kind of open forced-feed hot-water system that had given us excellent results for 15 years.

Fig. 3 shows a 600 k.w. steam turbine at the old rating. Directly under the turbine is shown its connections—simply an expansion joint and distance piece connected to a surface condenser. Under this is a wet air pump of large capacity. Also it will be noticed that the circulating pump draws water from the cold intake well. When not used for heating, this turbine runs at 28 in. of vacuum, drawing its water from this well and discharging it into the discharge tunnel under a siphon.

When heating is called for, the 12 in. valve into the intake tunnel is shut off, and the valve connected with the 178,000 sq. ft. of mill radiation is opened. As this system is full of water connected with an open expan-

sion tank condensing water from the return end of the heating system is drawn out and discharged into the inlet end of this system.

There is still 28 in. of vacuum left, but as this water is gradually heating up it will get down to 26 in. of vacuum if we continue to circulate this water and carry 600 k.w. on the turbine. Also our buildings will become warmer and the vacuum lower than necessary, but we would be losing as we would do under the same condition with steam heating. Here is where the economy and flexibility of heating under partial vacuum comes in.

It will be assumed that the heat due to 26 in. of vacuum meets the required outdoor temperature of 50 deg. fahr. and we want to hold this

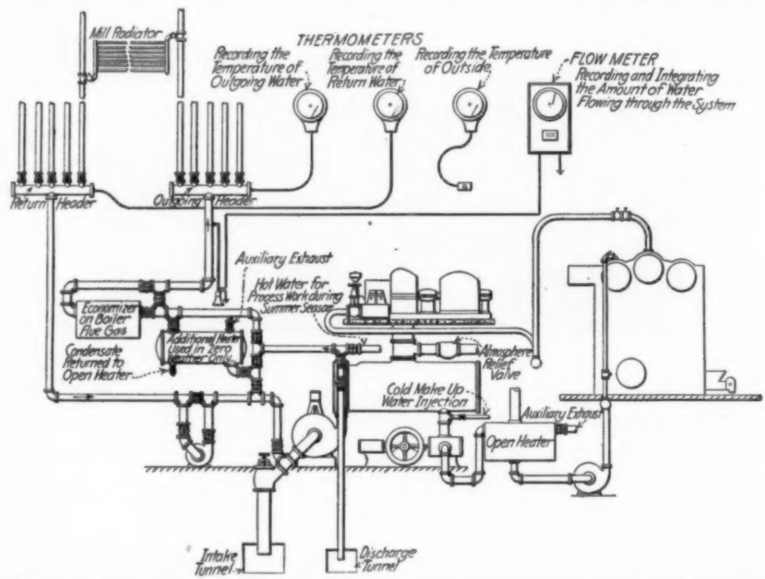


FIG. 3. ARRANGEMENT OF HEATING SYSTEM FOR CHASE METAL WORKS

vacuum and maintain about 120 deg. fahr. on the coils. This is done in two ways—one, by putting some of this 600 k.w. load on the other turbines; the other way, by running some cold injection water into the wet air pump as this water is used for boiler makeup. If there is an excess use for manufacturing purposes there is no loss, for the full 600 k.w. load is being kept up as well as carrying the excess heat back into the boilers, or for manufacturing purposes, and as the weather gets colder the same thing is done as explained in connection with the jet condenser, cut the amount of injection water to meet the weather requirements. It should be borne in mind that this wet vacuum pump becomes a jet condenser when water is put into it to maintain 26 in. of vacuum. This pump should be of a capacity to meet these requirements. At weather

about 20 deg. fahr. we find that this water can all be cut off and a bigger load is thrown on the turbine to meet colder weather conditions.

It should be noted here that the higher the ratio of pipe radiation to space is, the greater the economy of this system. The writer recently learned of a system put in of such ratio, that running under 24 in. of vacuum at the condenser would care for their coldest weather conditions. This on a steam engine would be supplying the heat practically for nothing. On a steam turbine we would be losing 4 in. of vacuum, which would call for more steam to go through the turbine, but as we use all the heat that goes through, our loss is simply the loss of capacity due to the amount of steam used.

We have also before this subdivided the 178,000 sq. ft. radiation. When the heating water was turned on to substitute the river water, if there was enough pipe radiation connected to this turbine to balance the heat from 600 k.w. load, it would not be necessary to put cold water in the wet vacuum pump, neither would it be possible to increase the heat on the system as it would be a fixture. So all the radiation of this mill can be put on this 600 k.w. turbine and balance the heat with the right load or the right amount of injection water.

In a short time the weather calls for the 600 k.w. load and no injection water. Then this 178,000 sq. ft. system can be run as stated above in one system. Next we will need more heat or less radiation, so we leave 130,000 sq. ft. of this 170,000 sq. ft. on this one turbine which balances the load during real cold weather and, as the other 48,000 sq. ft. is on the office and other buildings that require a temperature of 70 deg. while the 130,000 sq. ft. is on the mill which calls for 60 deg., this contributes to the economy of the system, very much the same as doubling the radiation to permit a low heat and a high vacuum. The economizer and the heater are shown in Fig. 2.

After the heating water is passed through the condenser, its heat is stepped up in the economizer, but during very cold weather, which is exceptional, a little exhaust from the steam auxiliaries is turned into this heater rather than lowering the vacuum.

Some years ago the writer wrote a paper showing a turbine running condensing with no waste of heat up "the stack," stack temperature being 260 deg. fahr., and no heat going to waste down the stream. The radiation at that time was about 75,000 sq. ft. and the load about 400 k.w., which enabled us to do it. The present load is about 8,000 k.w. In connection with this system the writer only claimed the credit for using the condensers directly as a substitute for the heater, making it possible to get the same economy and flexibility on a system on partial vacuum connected to turbines.

All that is necessary is to connect the hot-water system with the condenser tail pipe and regulate it as explained, which gives the simplest system in care and the most flexible one in economy. All the power will be gotten down to the vacuum that corresponds with the heat that is used, with the ability to follow all varying temperatures which is not possible with any other heating system.

Our steam pressure is 200 lb. with 1.40 deg. superheat. The trend is toward higher pressure. Assuming that a turbine runs at 300 lb. pressure with 150 deg. fahr., superheat and exhaust at 50 lb. and that it uses this exhaust steam for process work, usually heating water to 180 deg. fahr. and up, there is no reason to lose this power from 50 lb. pressure to 180 deg. of heat. This loss is equal to a turbine of the same size running at 50 lb. pressure and about 12 in. of vacuum.

TABLE 1.—OPERATORS CHART OF TEMPERATURES FOR HEATING WATER

Outside deg. fahr.	Outgoing Header deg. fahr.	Outside deg. fahr.	Outgoing Header deg. fahr.	Outside deg. fahr.	Outgoing Header deg. fahr.
70	100	46	125	22	160
68	102	44	128	20	162
66	104	42	131	18	164
64	106	40	134	16	166
62	108	38	137	14	169
60	110	36	140	12	172
58	112	34	143	10	175
56	114	32	146	8	178
54	116	30	149	6	181
52	118	28	152	4	184
50	120	26	155	2	187
48	122	24	158	0	190

NOTE—When weather is raw and windy run 5 deg. higher than chart calls for.

With the partial vacuum principle we get this extra power, for we start with our heat where our power leaves off. If we need a higher temperature in the water, "bleed" from some stage to make up this difference. The condensate at 180 deg. is returned into boilers. We cannot get all that there is in the steam exhausting at 50 lb. back pressure, unless we take all the power out down to the temperature we require.

In connection with heating buildings, the partial vacuum principle meets all weather requirements in its flexibility, enabling us to move with the weather and give the temperature required on the heating coils from zero, with about 8 in. of vacuum, to 60 deg. fahr. At that temperature it will be desirable to cut the vacuum from 28 in. to 27 in. to meet the 60 deg. fahr. outside temperature, and, at the same time obtain the full power of steam used due to the vacuum received.

When the writer took up the question with some good heating authorities of discarding the heater and suggested heating directly with the surface condensers directly from the full turbine load, he said that it would be necessary to sit beside the system 24 hours a day; the reverse, however, has been true, for it is simpler to manipulate than heating with what the writer terms live exhaust, or exhaust steam not connected under a vacuum.

The oiler at the plant cares for the heating system in connection with his other work. He first consults the outdoor recording thermometer and then looks at the chart marked "temperature on heating system." On his round he consults the temperature on the outgoing hot-water recording thermometer. This is adjusted by increasing or decreasing the load or by putting more or less cold water in the condensate pump. These charts

also enable us to get the proper credit and show if the proper amount of heat is being supplied.

For example: Charts for the week ending February 10, 1923, show on one of these systems, a load of 95,680 k.w. hr., and an average outdoor

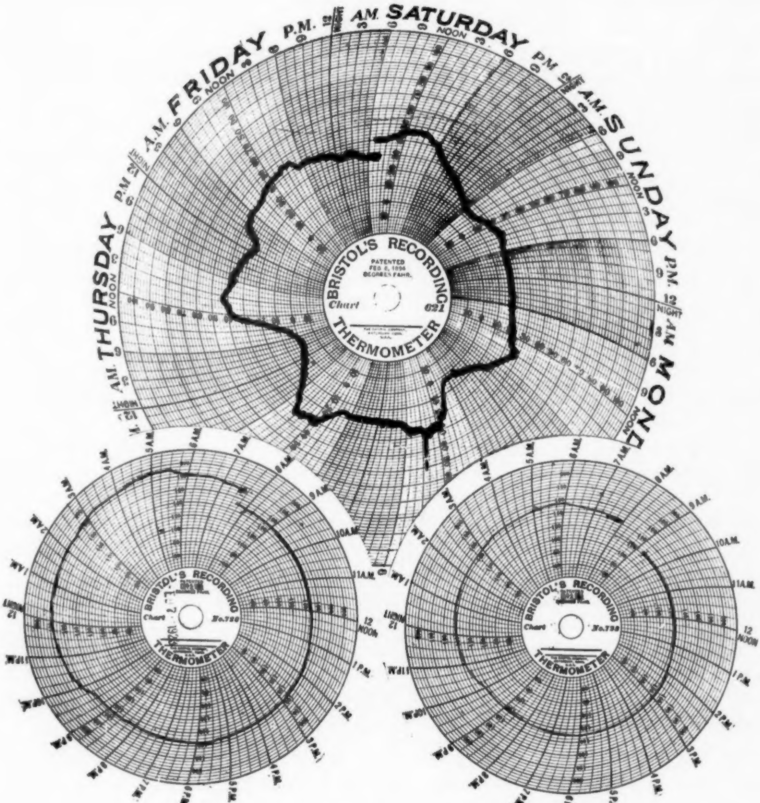


FIG. 4 (ABOVE). RECORD OF OUTSIDE TEMPERATURE DURING WEEK OF FEBRUARY 10, 1923

FIG. 5. TEMPERATURE OF OUTGOING WATER

FIG. 6. TEMPERATURE OF RETURN WATER

temperature of 25 deg. fahr., calling for 156 deg. fahr., on the outgoing hot water with a return drop of 26 deg. The flow meter chart shows 140,000,000 lb. of water sent through the system at 26 deg. fahr., difference in temperature. The outlet and return heat shows a radiation of 3,640,000,000 B.t.u. of which 2,800,000,000 B.t.u. come from the turbine and the remaining 840,000,000 B.t.u. come from the economizer; to

meet the extreme cold of this week live exhaust was admitted into the heater from steam compressors and steam pumps. These steam auxiliaries are used for this purpose and breakdown purposes only.

There is nothing special installed with this surface-condenser turbine equipment to provide for the heating feature. The writer would, however, recommend the jet or wet vacuum pump to be of a style that would act as a check. In other words, it should be an open-valve pump set below the condenser, so that if a tube ruptures when the equipment is shut down the water would run right through. In our case we must confess that this emergency has not been provided for. The reason for this is that we make our own tubes and never felt the necessity of it.

For high buildings the body of the condenser should of course be designed to meet the static head. In our case we have a pressure 40 lb. on the ordinary condensers.

No. 665

DISTRICT HEATING PROGRESS IN BOSTON

By D. S. Boyden[1], Boston, Mass.

Member

THE designation *central station heating system* is more or less ambiguous as it does not suggest any limitations to the area served, and gives the impression that heat is supplied more or less generally as in the case of water, gas or electricity. *District heating* is a more comprehensive designation as it suggests limits to the area served, and the public will assume that the district or precinct served with heat is not only fixed, but is less than that served with water, gas or electricity in the same territory.

Why is it that district heating is not more popular and more commercially used today in the larger cities? It is not because it is a new or untried servant, for steam and hot water were mediums of heating long before the introduction of our indispensable servant, electricity, as a commercial medium for light, heat, and power. Thus we may ask why—with our great water, gas and electric utilities, and the convenience they afford, do we not have district heating?

Does any utility in this latitude give greater convenience than district heat? Imagine for the moment a continuous and ample supply of heat in your home—thermostatically controlled—resulting in a constant house temperature with any desired ventilation, or, if you please, the coal and ashes coming in and going out the same pipe with no effort on your part except to pay a monthly charge. It is convenient. So it is necessary to look in other directions for the causes retarding the progress of this business.

SIX FACTORS RETARDING DISTRICT HEATING DEVELOPMENT

A review of the situation will reveal several factors which have in the past contributed to the present unsatisfactory status of the industry such as:

1. The joint operation of heating and electric plants in the early days.

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[1] Supt., Steam Heating Dept., Edison Elect. Ill. Co.

Paper presented at the Semi-Annual Meeting of American Society of Heating and Ventilating Engineers, Chicago, Ill., May, 1923.

2. Failure of hot water heating systems.
3. Lack of proper heat measuring apparatus.

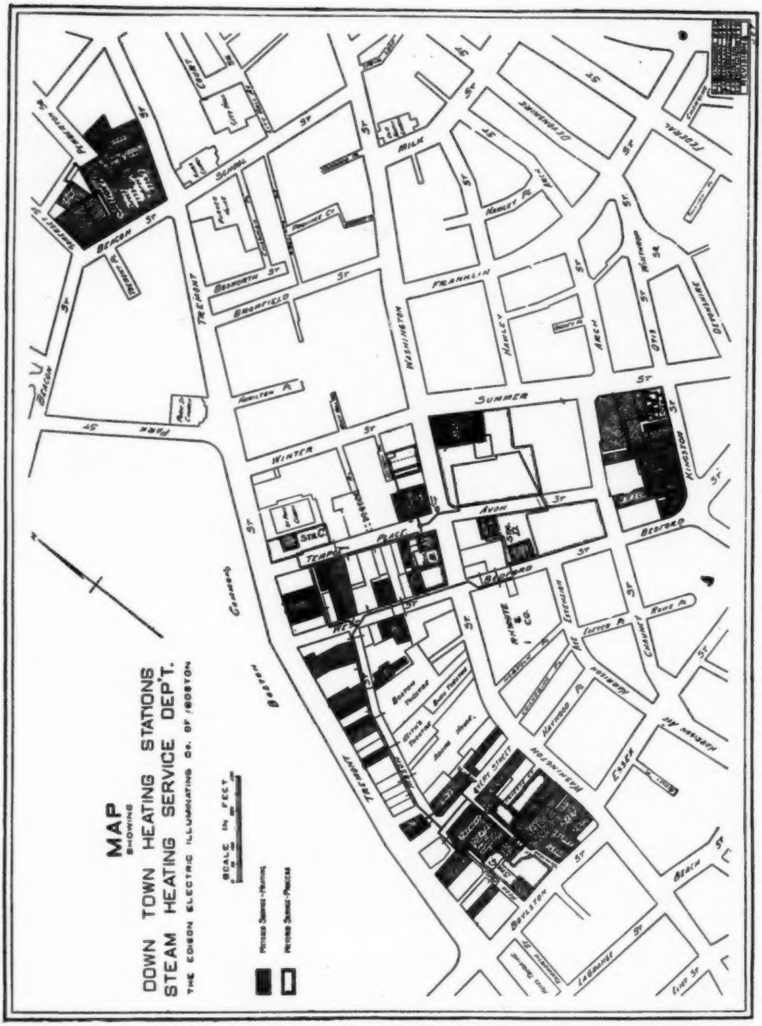


FIG. 1. DOWN TOWN STEAM SERVICE STATIONS AND MAINS OF THE EDISON ELECTRIC ILLUMINATING CO., BOSTON

4. Inefficient distribution construction.
5. Inadequate rates.
6. Lack of financial support.

It is hardly necessary to analyze these several factors, but it might be well to touch briefly on each to bring out its relative influence.

In the early days, the electric companies used steam engines of the reciprocating type to operate their electric generators and the exhaust steam from these engines was in many cases sold for heating, at a cost slightly above the loss incurred by operating non-condensing, plus an amount to cover the fixed charges on that portion of the investment used exclusively for the heating business. Many district heating systems were established under these conditions, and for a time operated at a satisfactory profit.

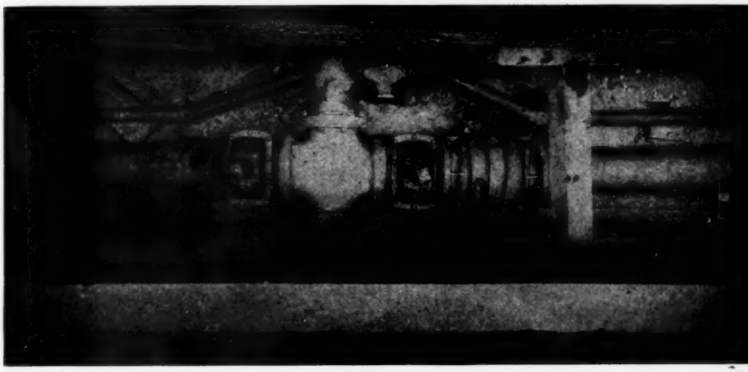


FIG. 2. MAN HOLE LOCATION DURING CONSTRUCTION

The rapid development of the electric business, together with long distance transmission of electric current, the introduction of steam turbines, the development by hydro-electric current, soon made it desirable to either relocate the electric plant or substitute transmission power for steam power. Immediately the heating business fixed in its location was obliged to carry the entire capital burden and forego the low cost of by-product steam which it had previously enjoyed, for in most cases no increase in rates could be effected.

Under these new conditions operations were highly unprofitable as might be expected. Hot water heating has contributed a negative impetus to the success of district heating due primarily to the greater weight transmitted per B.t.u., and the inability to measure the heat abstracted from the water by the consumer.

In 1887 a unique hot water heating system was built in Boston and undoubtedly was the worst failure, which ever came to the writer's attention. The scheme employed was what is known as forced circulation of hot water, the water being circulated at a temperature of 400 to 450 deg. fahr.

Some very interesting values are noted in comparing water and steam at these temperatures; for instance, 400 deg. fahr. represents 232 lb.

pressure and has a heat content per lb. of 374 B.t.u., while steam at 5 lb. pressure contains 1155 B.t.u. per lb. With the water at 450 deg. fahr., representing a pressure of 406 lb., it carries a heat content of 428 B.t.u. per lb. as compared with 1155 B.t.u. in 1 lb. of steam at 5 lb. pressure. Therefore, with water five to six times as much weight must be transmitted for the equivalent heat of steam. The system lasted but a few months. This failure destroyed capital confidence in the heating industry at a time when it was most needed. A detail story of the scheme will be found in the July 15, 1916, *Bulletin* of the *National District Heating Association.*

Steam until recent years has been sold on a flat rate method of charges, that is, so much per season or per month. In the case of atmospheric heating, it has been customary to charge so much per square foot of radiation per season; for steam used in process work, so much per hour or so much per month with fixed hours of use per week or per month.

All of us know too well the fate of vending water, gas or electricity, or any other commodity upon an unmeasured basis. The effort to save falls entirely upon the helpless vendor. Modern underground distribution systems of today as contrasted with the early systems show a great improvement; to mention but a single detail, the elimination of the screwed joints in pipes as compared with the welded joint practice of today. Drainage, insulation, and expansion have all been greatly improved.

Rates have been given as a retarding factor in the development of district heating. In the early days, charges were necessarily based on estimated use, or the flat rate method, there being no reliable meters, and with a rate structure based on exhaust steam, a by-product of another utility, and carrying no provision for excess cost of coal, was entirely inadequate to make the heating business successful.

The last and most important retarding factor today is that of financial support. This support is ever present and can be readily secured where the yield is good and the investment sound, but such conditions as have been here cited, have done much to discourage capital in district heating. There are today operating throughout the country a number of district heating companies of modern design, efficiently operated with adequate rates, carrying protective coal cost charges which show a fair return upon the invested capital. These systems will in time restore the investment confidence so essential to the development of district heating.

The recent world war and its attendant soaring costs of coal, labor, and materials precipitated an adjustment of heating rates throughout the country, and this has done much to clarify an unhealthy condition.

There is at this time an earnest and potential demand for district heating in congested business and residential districts of cities. This demand is the natural result of the universal use of electricity for all light and power requirements. The isolated plant so frequently installed in the past is now installed as the exception. Thus, the consumer reducing his mechanical operations to the production of steam for heating or process work desires to eliminate entirely his mechanical operations, and appeals to the central station for assistance or relief in this direction.

In Boston, Mass., steam has been supplied to properties adjoining the steam plants of the electric company as long ago as 1887, and since that time additional heating plants have been acquired so there are at the present time seven individual heating stations, ranging in capacity from 10,200 lb. of steam per hr. to 34,000 lb.

The total hourly capacity of these stations is approximately 150,000 lb., or sufficient to supply 600,000 sq. ft. of direct radiation, or capable of heating to 70 deg. fahr. in zero weather, buildings having a combined volume of 60,000,000 cu. ft.

The annual steam output of the seven stations has increased from 17,000,000 lb. in 1905 to 165,000,000 lb. in 1922. Six of these stations are located in an area having a radius of 500 ft., and four are within an area of a radius of 250 ft. Owing to the character of the service supplied by these several stations, it is necessary to keep them all in continous operation throughout the year, although during the summer or non-atmospheric heating period, the total demand on the combined system is slightly greater than the capacity of the smallest station. This condition with a growing demand for steam service, gave rise to a serious consideration of the possibilities to be realized by the interconnecting of certain stations by underground steam mains.

As a result of these considerations, the company has installed underground mains via Haymarket Pl., Avery, Mason, West and Bedford Sts., and Temple Pl., connecting five stations, thus bringing this group of stations into a single system. One station will be increased in boiler capacity from 14,000 lb. per hr. to approximately 56,000 lb. This is accomplished by installing an additional boiler and equipping all boilers with mechanical stokers, and providing the necessary mechanical draft and automatic control. This station under normal operating conditions will be the base plant for the interconnected system with the remaining stations serving as auxiliaries. The overall efficiency of the interconnected system will show a decided improvement over the previous method of operation, and in addition, the interconnecting tie lines will serve as a distributing system in the streets mentioned with a total length of approximately 2500 ft.

The installation of underground steam mains in a congested city street is not a simple problem, as it is necessary to keep the pipe at a predetermined grade, draining all low points, and the pipe must also be thoroughly insulated and water-proofed to preserve the structure and prevent undue waste of heat. These mains are installed at an approximate depth of 6 to 10 ft. below grade, depending upon the existing sub-structures and necessary drainage pitch. After the trench has been opened to the desired grade, approximately 4 ft. wide, a hollow tile under-drain is laid in crushed stone to take care of the drainage of surface waters. Upon this crushed stone, a foundation of 5 in. of concrete is laid and on this foundation, the pipe is installed with its necessary anchor supports and service connections, and where required, expansion joints are provided.

The pipe is genuine wrought iron, all joints made by the oxy-acetylene welding process. At all anchor fittings, the pipe is welded to steel flanges. The steam pipe is covered with two layers of asbestos insulation, having

a total thickness of 3 in. This is then covered with a 5-ply roofing paper which is cemented at the joints and then painted with two coats of asphaltum paint. The return line is insulated in a similar manner, but the insulation is necessarily not as thick, as the temperature of the condensation is very much lower than the temperature in the steam line. Both pipes are then enclosed in a multi-cell vitrified tile envelope and plastered with mortar to provide protection against seepage entering the conduit. The temperature on the outer surface of the enclosing envelope is slightly higher than the surrounding soil, so perfect is the insulation. At intervals the hollow tile voids are sealed with asbestos cement to

FIG. 3. TRENCH CONSTRUCTION

prevent transfer of heat by circulation. At all street intersections and other points where valves are installed, or where drainage of the steam main is required, suitable manholes are installed to receive the apparatus.

Expansion is taken care of by corrugated copper joints, corrugations being fitted with distance straps to limit the travel of the individual corrugations. The factor of safety to be employed per corrugation determines the width of the limit rings between corrugations. It is impossible to predetermine the proper pipe size of an underground main in a downtown city district, as the load distribution is constantly changing, therefore, these sizes are more or less arbitrarily selected. In Boston, 4 in. and 8 in. mains are the standard, the former being used on alleys and dead-end streets where the future loads can be approximately fixed; 8

in. mains are installed on all other streets or public alleys. The maximum allowable drop in steam pressure is maintained by reasonable increase of pressure at the source; further drops are taken care of by installing parallel lines on adjacent streets connecting them at street intersections. Booster feeder lines may be installed from the station to specific points in the network should the load exceed the capacity of the system mains. This practice gives the greatest flexibility for maintenance and at the same time provides a means for reducing the radiation losses during light load periods.

The steam business available from the distribution mains amounts to approximately 200,000,000 lb. per year. Already new business to the extent of approximately 50,000,000 lb. per year has been connected, and approximately 25,000,000 lb. of new business has been contracted for.

Numerous questions are asked us daily, seeking information on various phases of the steam heating service, which indicate a strong interest in the service and an apparent lack of knowledge as to how the service is rendered commercially.

RATES

Steam is sold under three agreements:

1. The seasonal agreement which provides for a discontinuance of steam service during the non-atmospheric heating period.

2. Yearly agreements, which cover the continuous supply of steam throughout the year.

3. Yearly guarantee agreements, which provide that the customer guarantee a minimum $120 per mo. each month of the year. The rates for steam are the same under each agreement, and are as follows:

STEAM SERVICE RATES

For the first	50,000	lb.	of	steam	per	mo.	$1.30	per	1000	lb.
For the next	50,000	"	"	"	"	"	1.20	"	"	"
For the next	100,000	"	"	"	"	"	1.00	"	"	"
For the next	300,000	"	"	"	"	"	.90	"	"	"
For the next	500,000	"	"	"	"	"	.80	"	"	"
For all over	1,000,000	"	"	"	"	"	.75	"	"	"

Under the yearly guarantee agreement, the customer is required to provide an approved pumping plant for the return of his condensation to the return line of the company, in which event an allowance is made of .05 per 1000 lb. of steam from the above rates.

In addition to the above rates, a coal charge is made when the cost of coal delivered to the steam heating service station exceeds $4.50 per long ton. The method of computing this charge takes into consideration the efficiency of plant operation, and any improvement tending to reduce the cost of steam sold is reflected in reducing the customer's cost as is here shown:

If in any month the average cost of coal used by the company in its steam service stations exceeds $4.50 per long ton in the bunkers, an additional charge will be made, computed as follows: The number of long tons of coal used for all steam service during the preceding 12 months will be divided by the number of thousand pounds of steam sold (whether measured by meter or estimated) during the same period; the quotient being the average amount of coal required for each 1000 lb. of steam sold, will then be multiplied by the cost of coal in excess of $4.50 per long ton for the month in question, and the product will be the additional charge per 1000 lb. of steam.

Steam is transmitted at approximately 95 lb. gauge pressure and delivered to consumers through reducing valves furnished by the company at

FIG. 4. VIEW OF TYPICAL UNDERGROUND CONSTRUCTION

either 15 or 40 lb. gauge pressure, the pressure supplied being determined by the company. Service pipes to customers are either 2½ in., 4 in. or 6 in. as required. The pipe is extra strong and terminates in an extra heavy gate valve of the inside yoke rising spindle type. The reducing valve embodies in its design special protective features. With no steam flowing, the spring acts against the diaphragm keeping the valve closed. To put the valve into operation, the by-pass must be opened, allowing the initial pressure steam to flow into the low pressure side of the system and also into the diaphragm chamber. When the pressure builds up, it will act against the diaphragm, opening the valve, and allowing steam to flow through it. When the reducing valve has been opened, and functioning, the by-pass valve is closed, allowing steam to pass through the reducing valve only.

Thus, if the pressure is lost in the diaphragm chamber, the spring automatically closes the valve, also if the customer should close the company's service valve, the reducing valve would be closed until opened in the manner described. As service and by-pass valves are sealed, a company representative must make the restoration. In event of a failure in the service valve requiring repairs, a section of the underground mains would have to be cut off, entailing an interruption of service; therefore, it is desirable to prolong the life of the service valve by using only in emergency cases. With the bursting of the diaphragm or pressure feeling pipe, the reducing valve automatically closes. Each reducing valve is made up as a standard fixture, irrespective of size. A pressure indicating gauge is attached to the pressure feeding pipe which has a metal disc directly under the glass, a V-notch is cut in this through which the pressure if within 5 lb. above or below normal can be read. With the customer's stop valve closed and the reducing valve leaking, the pressure will go up and the pressure gauge hand disappear.

The quantity of steam supplied to consumers is measured by meters recognizing that a pound of steam may be a pound of water evaporated into steam or a pound of steam condensed into water. Therefore, it is possible to measure steam by either one or two methods: *First,* measure the flow at the point of supply, or *second,* the condensation at the point of discharge. Flow meters function on the pitot tube, or orifice principle, the measurement being dependent upon the drop in pressure through the tube or orifice. This drop in pressure through mechanical transmission is transcribed in pounds on a chart driven by a clock movement and at the same time integrated on a dial index, very much like an electric meter. Flow meters are used in large installations or where steam is supplied to open jet appliances. Condensation meters are more simple, being either automatic weighing or volumetric devices, operating on the revolving or tilting bucket principle, the rotary or tilting action connecting to a dial index designed to record the pounds of steam passing.

Where condensation meters are used, the condensation from the heating system or appliances must pass through indirect radiation before entering the meter. This is necessary for three reasons: *First,* to obtain a uniform temperature of the condensation for weighing purposes; *second,* to enable the customer to extract the maximum amount of heat from the steam he has purchased; *third,* to insure against the discharge of excessively hot water to waste or sewer. The condensation of customers taking steam under the yearly guarantee agreement is discharged into the company's return line.

On the inlet to all condensation meters, a standard semaphore is installed which displays a signal when the temperature of the condensation has exceeded 130 deg. fahr. A further signal is displayed when the temperature of the condensate has exceeded 175 deg. fahr. Frequent indications of these temperatures being excessive, cause the necessary action to be taken with the customer to place his apparatus in proper condition to avoid these excessive temperatures. Semaphores are also installed on discharge lines of customers' approved pumping plants where the condensation is being returned to the company's return mains. The

observations of this semaphore indicate when the condensate has been returned at an abnormally low temperature. All customers' reducing valves, semaphores, and meters are read bi-weekly during the period which service is supplied.

Where flow meters are used and the steam is maintained at a constant pressure, the meter accuracy compares favorably with the electric meter. This is also true of condensation meter when the condensation is maintained at a temperature not greater than 130 deg. fahr.

HOW DOES THE COST OF PURCHASED STEAM COMPARE WITH PRIVATE PLANT OPERATION?

This question is probably raised more frequently than any other question and, obviously, it would be impossible to answer the question with any degree of accuracy without a careful analysis of the operating costs and efficiency at which the particular private plant made steam. But to give some idea of the comparative cost, small heating plants in buildings having a volume of less than 500,000 cu. ft. of space, develop approximately 4 to 5 lb. of steam per lb. of coal purchased, or—roughly, 8000 to 10,000 lb. of steam per ton of coal. As a comparison, a ton of anthracite coal making 10,000 lb. of steam and costing $15.75, would give an average cost of $1.57 per 1000 lb. of steam generated.

Referring to the steam service rates, it will be observed that the maximum rate for the first 50,000 lb. of steam per month would be $1.30 per 1000 lb.; to this must be added the coal charge which at present cost of coal is $.24, making the price per 1000 lb. for purchased steam $1.54, or a saving of $.03 per 1000 lb. in cost of coal alone. As the amount of steam purchased increases, the rate per 1000 lb. is reduced, while the cost for coal would remain substantially the same irrespective of the amount purchased in buildings of this size. There are many other items of costs entering into the private plant operation greatly increasing this differential.

The most important factor affecting the comparative cost of purchased steam vs. private plant operation, is the customer's utilization of the service. In the small private plant, labor is required to stoke the boiler, and when coal is used as a fuel, the task is frequently an arduous one with the hours of stoking increasing with the lower ambient temperatures. Any relaxation upon the part of the stoker, or any deficiency in the amount of heat supplied at least saves fuel for the owner. With a district heating service, the conditions are quite the opposite, the convenience of an unlimited supply of steam insures continuous and efficient heating irrespective of the temperature, and unless a vigilant watch be kept upon the steam control, an unnecessary waste will occur, resulting in an excessive consumption and cost.

The district heating company sells a service which is incomparable to the small isolated plant service, and it falls upon the heating company to educate the consumer to operate this service economically.

WHAT ARE THE ADVANTAGES OF PURCHASED STEAM SERVICE?

Just a few of these advantages will be enumerated here. Purchased steam from a central source not only has the fundamental advantage of

convenience but it is a decided step in civic improvement. To enumerate:

1. Civic Improvement:
    a. Eliminates smoke nuisance.
    b. Keeps coal trucks off the street.
    c. Reduces dust in the street.
    d. Keeps ash trucks off the street.
    e. Keeps coal and ash away from customers' buildings.
    f. Improves traffic on streets and sidewalk.
    g. Eliminates boiler explosions in buildings.

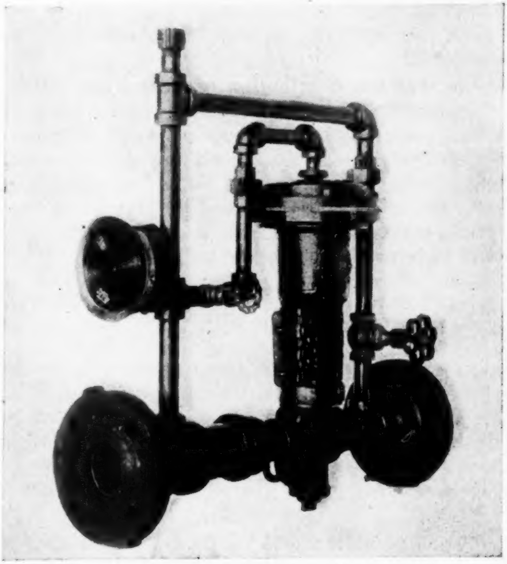


FIG. 5. STANDARD REDUCING PRESSURE VALVE FIXTURE

    h. Reduces fire hazard.
    i. Removes gas fumes from buildings.
    j. Makes the city safer.

2. Greater Convenience:
    a. Heat available when wanted at any hour, day or night.
    b. Turn on valve vs. building of fires.
    c. Saves time for owner, manager, or building superintendent.
    d. No boiler to buy or operate.
    e. No firemen to employ.
    f. No boiler inspection or insurance.
    g. Saves time of employees.
    h. No bookkeeping of coal or ash handling costs.

i. Gives the advantage of metered service.
j. Know every day, every hour, exact amount of heat being used.
k. Total cost of heat in one statement, monthly or weekly.
l. No tools to purchase or repair.

It is with these thoughts in mind, as well as the good business aspect, that we have proceeded with our expansion plan in Boston, and we believe we are promoting a service which will steadily gain in popularity with our larger customers.

In conclusion, the writer wishes to say, that the failures of many heating plants in the past do not indicate that district heating cannot be made to pay, but that conservative estimates, suitable rates and adequate financing are necessary.

It is probable that the distribution of heat from district plants will become more general following the tendency indicated in the supply of electric light and power and other public service products. Influences which might well bring about this result are the convenience and certainty of service, the freedom from labor complications, occasional scarcity of fuel, and a possible increased differential in the cost of fuel between the prices which will have to be paid by the district company and those which will have to be paid by the individual.

DISCUSSION

J. H. WALKER (written): This paper contains a commendable summary of the present status of district heating and should help to clarify the somewhat confused impression which seems to exist regarding this important industry. Most of the unfavorable influences which the author enumerates have now disappeared and the next few years should demonstrate conclusively whether or not district heating shall become a standard utility in our cities.

I would add one point to those which the author cites as having retarded the development of district heating and that is the *close competition* which exists between the isolated heating plant and the supply from the central station—a much closer competition than exists in the case of any other utility. In other words, a building owner will entertain the idea of operating his own boiler plant when he will not consider the production of his own electricity, or the pumping of his water supply. With the progress of our civilization, however, this condition is steadily changing and the consumer is becoming more appreciative of and more willing to pay for the advantages of central heating service just as he is now willing to pay for electricity for lighting in preference to gas or kerosene which were quite acceptable a few years ago. This is borne out by the very pronounced and increasing demand for heating service in those cities where it is being satisfactorily rendered, particularly in business districts.

In our residential districts there is little probability of the universal development of the service in its present form. The cost of underground mains is too great for the limited amounts of heat required and the lack

of permanence, in many cities, of the once high class residential districts makes the investment unattractive. I look rather for the perfection of better means of generating heat locally as is illustrated by the development of oil burners.

A noteworthy feature of the Boston development is the use of high pressure steam. As the author points out, the present tendency in district heating is toward the use of live steam which has not been passed through electrical generating units. Although this appears to be a wasteful practice, it is often the most expedient from a broad commercial standpoint, for the much greater investment in underground mains required when low pressure exhaust steam is distributed, together with certain other factors, may completely outweigh the saving in coal which might be made. The future, however, may witness a reversal of this practice if the relation between coal costs and investment costs changes considerably.

No. 666

REGISTER TEMPERATURES IN WARM-AIR HEATING

By V. S. Day[1], Urbana, Ill.

Non-Member

IT has been suggested by many persons who are interested in codes and formulae for the design and installation of re-circulating gravity warm-air heating systems that a maximum allowable air temperature at the registers of 140 deg. be specified. This specification would mean that when the thermometer indicated an outside temperature of 0 deg. the temperatures at the register faces should not exceed 140 deg. Such a condition is not impossible of attainment in a building of excellent construction in which nice attention to weather proofing has been given.

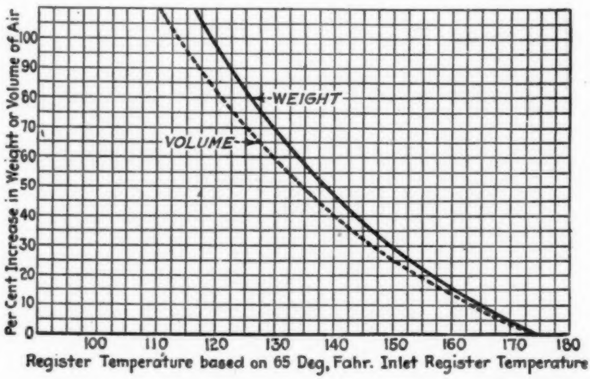


FIG. 1. CURVES SHOWING INCREASE IN WEIGHT OR IN VOLUME OF AIR TO GIVE CONSTANT HEATING EFFECT FOR VARIOUS REGISTER TEMPERATURES

[1] Assistant Professor of Research, Dept. of Mech. Eng., University of Illinois.

Paper presented at the Semi-Annual Meeting of American Society of Heating and Ventilating Engineers, Chicago, Ill., May, 1923.

In a building of average construction, a design based on such a low temperature would involve the use of much larger pipes and fittings than are used in plants operating at the more commonly accepted standard of 175 deg. This conclusion is based on the fact that to produce a given heating effect a greater quantity of air must be circulated at 140 deg. than at 175 deg. The accompanying curves and tables have been prepared to show the percentage increase in air quantities required to produce a common heating effect in terms of the quantity required at 175 deg.

The expression for the heating effect is as follows:

$$H = W \times 0.24 \times (T - 65)$$

in which

W = weight of air circulating at 175 deg., lb. per hr.
0.24 = fraction of a B.t.u. required to raise 1 lb. of air 1 deg. fahr.
T = register temperature, deg. fahr., = 175 deg.
65 = the temperature of the air at the return register, assumed to be the minimum for comfort near the return register.

If H is to be a fixed value, various values may be substituted for the register temperature, T, the expression may be solved for the corresponding value of W, the weight of air.

This process has been carried out in arriving at the values in Table 1 in which W is the weight of air required at 175 deg. fahr. The percentage increase in W is shown for various register temperatures including 140 deg. These relative values of W are shown graphically in the *weight* curve of Fig. 1. It may be observed that at 140 deg. at the register 46.7 per cent more air is required than at 175 deg.

TABLE 1. EFFECT OF VARIOUS REGISTER TEMPERATURES UPON THE WEIGHT OF AIR REQUIRED TO GIVE A CONSTANT HEATING EFFECT

Register T, deg. fahr	175	160	150	140	130
Rise (Reg. T — Inlet T)	110	95	85	75	65
Weight Air in Terms of Weight at 175 deg. fahr	W	$1.158W$	$1.294W$	$1.467W$	$1.692W$
Per cent Increase over Weight at 175 deg. fahr.		15.8	29.4	46.7	69.2

The values given in Table 1, line 3, represent relative velocities of flow in the return ducts. In the case of the warm air pipes and stacks, however, a correction for the density of the warm air must be made if velocity comparisons are to be made. Table 2 gives these results of Table 1 corrected for the densities, and the values are shown graphically in the *volume* curve of Fig. 1.

TABLE 2. EFFECT OF VARIOUS REGISTER TEMPERATURES UPON THE VOLUME OF AIR REQUIRED AT THE REGISTERS TO GIVE A CONSTANT HEATING EFFECT

Register T deg. fahr	175	160	150	140	130
Air Density	0.0624	0.0637	0.0646	0.0655	0.0665
Volume of Air in Terms of Volume at 175 deg. fahr	100	1.134	1.250	1.397	1.588

Per cent Increase over volume at 175 deg. fahr.	0	13.4	25.0	39.7	58.8

The curve shows that as the register temperatures are decreased the volumes required increase at a greater rate. At the suggested temperatures of 140 deg., the volume of air required would be 39.7 per cent greater than at 175 deg. This means larger stacks and fittings and 6 in. wall studding rather than 2 x 4 in. studding.

Just how much the increases might be has been determined in tests. A typical case follows:

A 3 x 10 in. second floor wall stack delivered 8000 B.t.u. per hour to a room, at 175 deg. register temperature. In so doing it handled 300 lb. of air per hour. To give a heating effect of 8000 B.t.u. per hour at 140 deg. the amount of air to be handled is (from Table 1) 300 x 1.467, or 440 lb. per hour. From the tests referred to it may be observed that the stack which handled 440 lb. of air at 140 deg. was between the sizes 5 x 12 and 5½ x 13, or approximately 65 sq. in. This corresponds to a stack 5 x 13 in. Such a stack could not, of course, be used without considerable extra expense in building and warm air installation cost.

On the basis of the arguments presented, it would appear impracticable to design warm air installations on a temperature at the registers of 140 deg. fahr., unless objections to the use of 6 in. studding are overcome.

No. 667

WALL STACK SIZE AND HEATING EFFECT AT THE REGISTER

By V. S. Day[1], Urbana, Ill.

Non-Member

As a part of the investigation of warm air furnace heating, being conducted by the Engineering Experiment Station of the University of Illinois co-operating with the *National Warm Air Heating and Ventilating Association,* an elaborate series of tests of warm air conductor pipes and fittings is in progress. The results obtained in this particular branch of the investigation present much information of value to the heating industry. The data contained in this paper are by no means a complete exhibit of results obtained, but are intended to give in a general way, results bearing on two topics which have occupied the attention of the Committee on the Warm Air Furnace Installation Code, as follows:

1. The proper ratio between stack leader pipe areas.
2. The relation between air temperature at the register and the heating effect available at the registers with various sizes of stacks and leaders. The application of the results to actual rooms is shown.

TESTING APPARATUS AND METHODS

The principal features and details of the electrically heated furnace used in the tests are shown in Fig. 1. This furnace, insulated with both an air space and magnesia, and having only one outlet, was designed especially for tests of the various warm-air accessories.

The anemometer is mounted permanently in the center of the inlet pipe. It is read through a celluloid plate in the pipe and has an external release for the clutch. For the purpose of calibration the inlet pipe may be detached from the furnace and connected to the standard air weighing calibration plant. It was found advisable to surround the inlet pipe with a screen in order to prevent draughts affecting the anemometer. The calibration curve for the anemometer is shown in the figure.

[1] Research Assistant Professor, University of Illinois.

Paper presented at the Semi-Annual Meeting of American Society of Heating and Ventilating Engineers, Chicago, Ill., May, 1923.

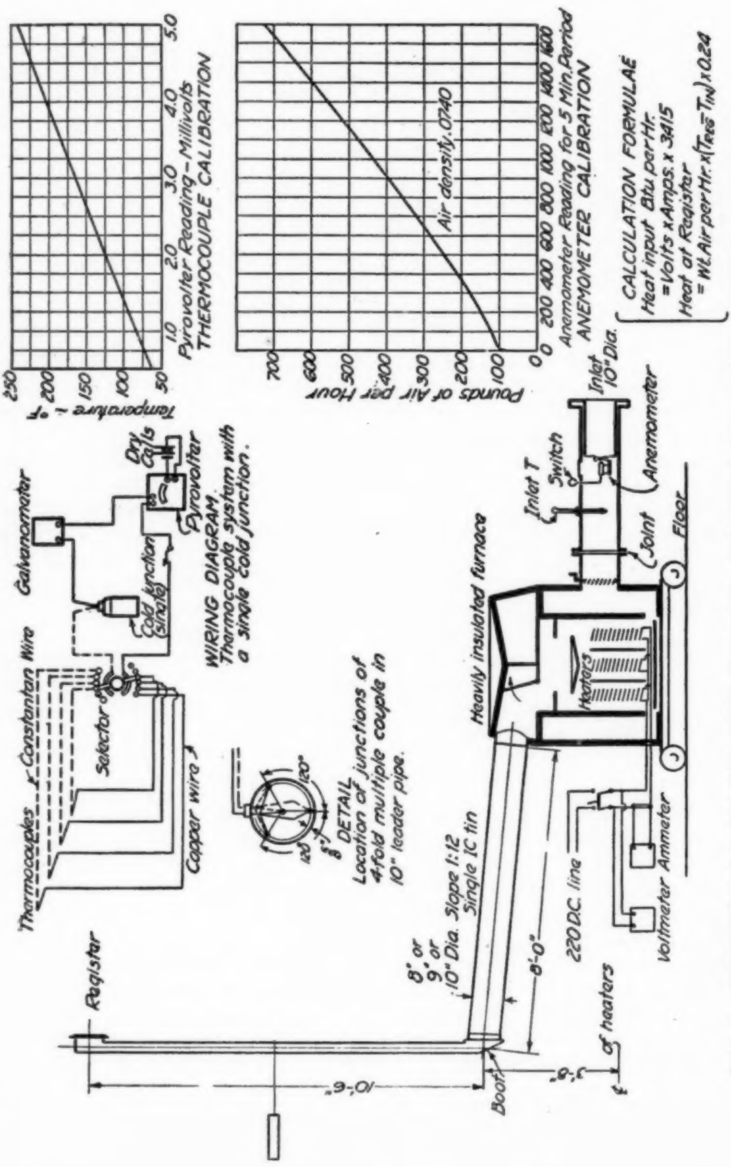


FIG. 1. ELECTRICALLY HEATED TEST FURNACE ARRANGED FOR STACK-LEADER RATIO TESTS

The thermocouple system for the test plant has two distinctive features, a single cold junction for the entire group of thermocouples, and multiple junctions for approximating the mean temperature rather than the temperature at a single point in the air stream. The wiring diagram for the thermocouple system is shown in Fig. 1, as well as a detail sketch showing the location of each of the four junctions of a multiple couple. A calibration curve for a thermocouple is also shown.

The use of electrical heaters operating on direct current with voltage control and regulation greatly simplifies the determination of the heat input to the furnace. The process of calculation involved is shown.

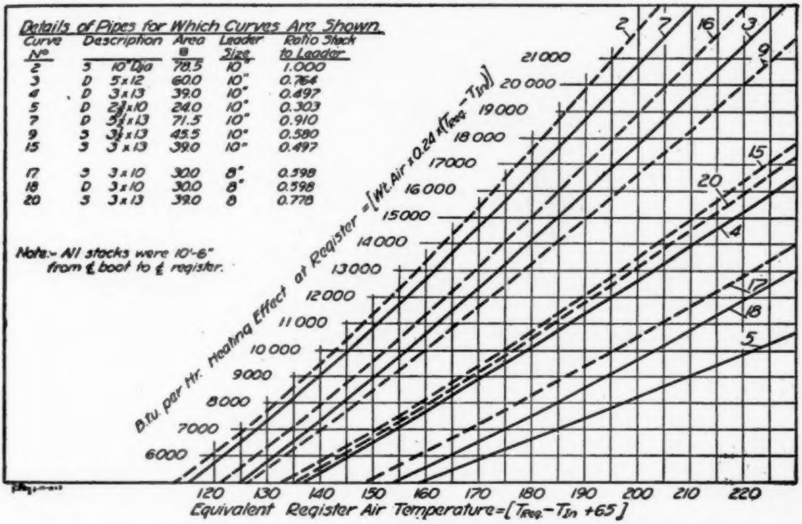


FIG. 2. RELATION BETWEEN AIR TEMPERATURE AT REGISTER AND HEATING EFFECT AT REGISTER FOR SEVERAL STACKS
Not all of the Stacks Tested are Included in This Set of Curves

Readings of the anemometer, the electrical input and temperatures at bonnet, boot entrance, boot exit, register throat, and entering air were made as nearly simultaneously as possible. From the quantities thus determined all the results were calculated. These computations were as follows:

$$H = (TR - T_1)\ W \times 0.24$$

H = heating effect at registers, B.t.u. per hr.
TR = air temperature at registers, deg. fahr.
T_1 = air temperature at inlet, deg. fahr.
W = weight of air, lb. per hr.
0.24 = specific heat of air.

$$I = V \times A \times 3.415$$

I = input to furnace B.t.u. per hr., gross.

V = volts.
A = amperes.
3.415 = conversion factor electrical to heat units.

The results of these computations were plotted in two ways; first, as heating effect against register air temperature, and second, as heating effect against gross heat input to the furnace. From these two kinds of curves all of the conclusions were drawn. A typical set of curves showing the relation between register air temperature and heating effect is shown in Fig. 2.

PROPER RATIO BETWEEN WALL STACK AREA AND LEADER AREA

In order to study the effect of various stack-leader ratios, twelve stacks covering a wide range of commercial sizes were tested. Of this number six were of single wall and six of double wall construction. The stacks were of uniform height (10 ft. 6 in.) and were connected to a 10 in. diameter leader 8 ft. in length. Details of these stacks are shown in Fig. 3.

Stack N°	2	3	4	5	6	7	9	10	11	13	15	16
Relative Size												
Single or Double	S	D	D	D	D	D	S	S	S	D	S	S
Size, internal	10"	5x12	3x13	2⅜x10	3x10	5½x13	3½x13	3x12	3x10	10"	3x13	5x12
Area Sq. in.	78.5	60	39	23.7	30	71.5	45.5	36	30	78.5	39	60
Ratio Stack÷Lead	1.00	0.76	0.50	0.30	0.38	0.91	0.58	0.46	0.38	1.00	0.50	0.76

FIG. 3. RELATIVE SIZE OF WALL STACKS TESTED TO DETERMINE RELATION BETWEEN STACK-LEADER RATIO AND HEATING EFFECT IN GRAVITY SYSTEMS

Single and double 10 in. diameter round stacks, numbers 2 and 13 in Fig. 3 were used as the basis of comparison in their respective classes. It was assumed that no stack larger in cross-sectional area than the leader pipe to which it connects would be used and none were therefore tested.

For these twelve stacks tests were run and curves plotted, as described above. In Fig. 2, numbers 2, 3, 4, 5, 7, 9 and 15 were a part of the series.

From the register air temperature—heating effect curves values were selected at three register temperatures and an average for each curve determined. The averages for the single stacks were then compared with the average for the 10 in. round single stack and the averages for the double with the 10 in. round double stack and expressed as percentages. When plotted these comparisons give the curves of Fig. 4. The curves of Fig. 4, therefore, show the relative values of the stacks for a *fixed register air temperature*. For convenience the corresponding heat inputs to the furnace were plotted.

Average values of heating effect were determined from the heat input—heating effect curves in the same general manner and the comparison of the average values made with the values for the best, or round stacks. The comparisons were plotted in Fig. 5 and since the values were based on the same heat inputs the curve Fig. 5 shows the relative value of the

stacks for a *fixed heat input to the furnace.* For convenience the corresponding register air temperatures were plotted.

From Fig. 4 it may be observed that the heating effect varies almost directly with the cross-sectional area of the stack. That is, a stack of 0.6 of the area of the full round stack delivers approximately 60 per cent of the heat for the *same register air temperature.* The single stack points fall along a line slightly above the double stack points, but tests made on single stacks tightly enclosed in plaster studding spaces show that the line representing single tin stacks falls very close to the line

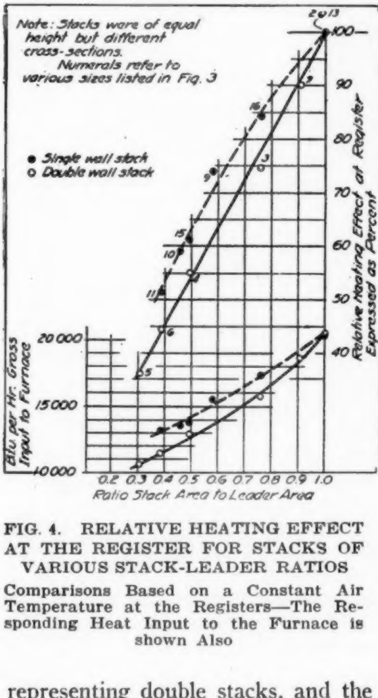


FIG. 4. RELATIVE HEATING EFFECT AT THE REGISTER FOR STACKS OF VARIOUS STACK-LEADER RATIOS

Comparisons Based on a Constant Air Temperature at the Registers—The Responding Heat Input to the Furnace is shown Also

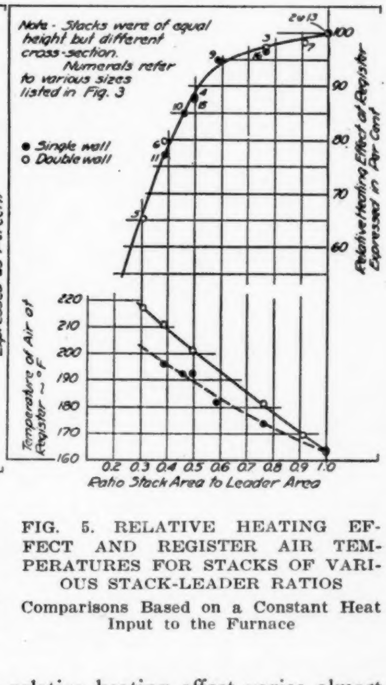


FIG. 5. RELATIVE HEATING EFFECT AND REGISTER AIR TEMPERATURES FOR STACKS OF VARIOUS STACK-LEADER RATIOS

Comparisons Based on a Constant Heat Input to the Furnace

representing double stacks, and the relative heating effect varies almost directly as the ratio of stack to leader.

From Fig. 5 another conclusion may be drawn, however. In this case, for a *fixed heat developed in the furnace,* the line is curved sharply. A stack only half the size of the full round stack delivers 0.9 of the heating effect at the register of the latter. It should be observed, however, that the temperature for the half size stack is over 190 deg. fahr., whereas the full stack delivers its heated air at 152 deg. fahr. The ability of the small stack to deliver so effectively, on the fixed heat input basis, is the result of lower transmission losses from the furnace casing, and higher furnace efficiency.

The difference between Figs. 4 and 5 introduces a nice point for discussion. On the basis of Fig. 5 it would appear not worth while to use stacks larger than 0.80 of the area of the leader pipe. In order to understand the curves thoroughly it must be observed that Fig. 4 expresses the design condition and Fig. 5 the operation condition. A correct design must be based on a *uniform and reasonable* register air temperature throughout. In operation the heat input is controlled and the register temperatures vary with the size of the various stacks installed, and in order to give the required heating effect the temperatures may become very high. This would be an unsatisfactory condition.

Fig. 5 shows that the advantage of using large stack-leader ratios lies in the reduction of register air temperatures for it shows that no ap-

Coefficients for temperature range 0° outside to 70° inside:-
For wall 17 Btu. per Hr. per sq ft., For glass 80 Btu. ϕ Hr., For crack 84 Btu. per ft. per Hr.

Room	Description			Net			Heat Loss Btu. per Hr.				At 175° Reg.	
	Size	Windows	Exposures	Wall	Glass	Crack	Wall	Glass	Infilt.	Total	Stack	Leader
1	12½x11½x8	2 30"x60"	2 ½ Ceiling	239	25	35	4060	2000	2940	9000	3"x12"	8"
2	10x10x9	3 30x60	2 ½ Ceiling	193	37.5	52.5	3280	3000	4410	10690	3½x13	9
3	12x15x8	5 24x60	2	166	50	80	2820	4000	6720	13540	5x12	10
4	10x12x8	2 36x60	2 ½ Ceiling	206	30	38	3500	2400	3190	9090	3x13	9
5	12x12x8	2 36x60	2 ½ Ceiling	234	30	38	3980	2400	3190	9570	3x13	9
6	10x9½x8	2 24x60	1	75	20	32	1270	1600	2690	5560	3x10	8
7	6x8x8	1 24x24	1 ½ Ceiling	44	4	8	750	320	672	1742	3x10	8
8	10½x13x8	2 30x60	2	163	25	35	2780	2000	2940	7720	3x10	8
9	8x13x9	3 24x60	3 ½ Ceiling	283	30	48	4810	2400	4032	11242	3½x13	10
10	9x13x8½	5 24x60	2 ½ Ceiling	195	50	80	3320	4000	6720	14040	5x12	10

FIG. 6. SELECTION OF STACKS AND LEADER PIPES FOR TYPICAL ROOMS

preciable increase in heating effect accompanies a big increase in stack-leader ratio, above 0.6. It would appear best, therefore, to use that ratio which would give a *reasonably low* register air temperature, but at the same time will lie above the ratio 0.5 at which the curve breaks. Such a selection would be a value of 0.7 for leader-stack ratio and a temperature corresponding to 180 deg.

Other tests have been made to show the effect of varying the height of wall stack on this problem, and tests are to be made on various lengths of leader pipes. The results discussed were obtained on a stack and leader of average height and length.

RELATION BETWEEN AIR TEMPERATURES AT THE REGISTERS AND HEATING EFFECT FOR VARIOUS SIZES OF LEADERS AND STACKS

In Fig. 2 are shown curves for several sizes of wall stack on two leader pipe sizes. These curves do not by any means represent the whole of the tests. Not all of the sizes which have been tested are shown and additional sizes on 8 in., 9 in., and 12 in. leaders are to be tested. When

completed this group of curves will cover the whole range of commercial sizes.

The curves of Fig. 2 have a very practical application in the design of gravity warm air installation. Given a room to be heated and a predetermined register air temperature, it is only necessary to refer to the curves and select the wall stack size corresponding to the calculated heat loss and the required register air temperature. The curves given cover a wide range of stack sizes and room heat losses and when completed will cover almost any condition to be found in second story heating.

It may be observed that any room requiring 12,500 B.t.u. per hr. to make good its heat losses may be heated to 70 deg. on a zero day with stack which can be constructed in 4 in. wall studding space provided air temperatures of 175 deg. at the register are used. If higher temperatures are permitted smaller stacks may be used, or larger rooms may be heated with a given stack.

The class of rooms which may be heated with stacks in 4 in. studding at moderate register temperatures is by no means confined to small bathrooms. In Fig. 6 are shown the heat loss data for ten rooms taken from actual second-floor plans. These rooms were selected from drawings as being typical. The table shows the stack size required to heat the rooms to 70 deg. on a zero day with air leaving the register faces at 175 deg. fahr. The coefficients used in calculating the losses are shown with the table.

Of the rooms selected, eight can be heated with single stacks in 4 in. studding and if temperatures are permitted to increase to 190 deg. rooms larger than any of those listed can be heated with one stack in a 4 in. studding space. In case very large rooms, or rooms having very great heat losses are found they may be heated by using two relatively small stacks. Two such cases are shown in Fig. 3.

The advantage in large stacks lies in their ability to deliver air to the rooms at moderate temperatures. It is difficult, however, to justify an argument for temperatures lower than 175 deg. at the registers on a zero day. If a temperature of 175 or 180 deg. is adopted as a standard the tests show that the required heat for average second floor rooms can be delivered in stacks in 4 in. studding, provided no irregular conditions to interfere with flow, are introduced.

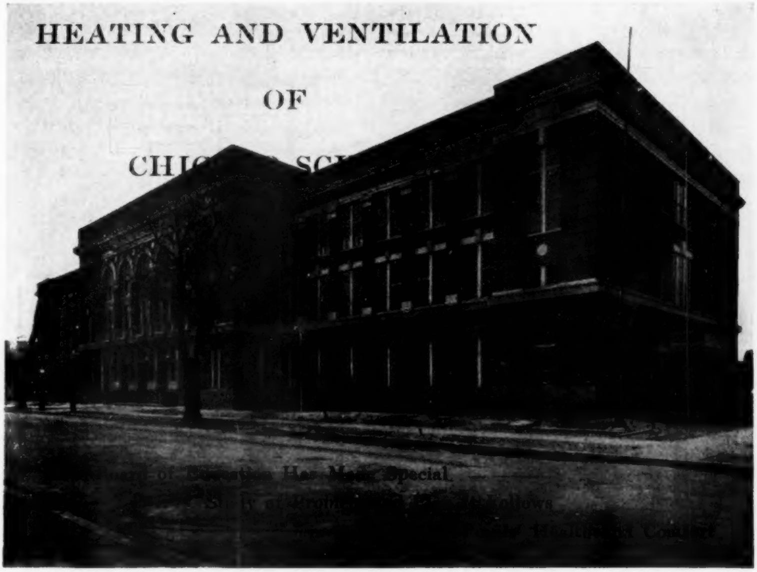
DISCUSSION

PROFESSOR DAY: The air temperature was about 175 deg. under average operating conditions. The stack is a factor, however, for, if it is small, the air temperature is very high. We obtained no such temperatures in our work.

E. B. LANGENBERG: This research work at the University of Illinois has given us some very practical data and it is being used in warm air heating practice today. For instance, when the wall stack tests had been completed a number of manufacturers closed up the air space between the inner and outer casing of their double style. This has borne out my

contention that single pipe is more efficient than a double pipe because of its greater area. There are certain things common to the research work at Urbana and at the Society's Laboratory at Pittsburgh and this work, when used in conjunction with the practical experience of all factors in the heating industry, will give a furnace code or one for any other phase of heating that will represent best practice.

PROFESSOR WILLARD: The data of Professor Day should be of particular interest to the sub-committee working on the Warm-Air Furnace Section of the Code, especially that part relating to warm air furnace performance. Furnace manufacturers generally do not recognize that a given furnace—and this is also true of a given boiler, has a variety of ratings depending upon the available draft and upon the register air temperature in the case of a furnace. For instance a furnace of a given diameter and arrangement of castings may be selected and designed for temperature of 150 deg. The capacity of that particular furnace might be increased from 60 to 100 per cent if the draft were available for a greatly increased combustion rate. The efficiency of the furnace or boiler will normally fall off somewhat in doing this, so it can be readily seen that the question of the effect of register air temperature on furnace rating or capacity is a vital one that should be considered by the Code Committee.



BY JOHN HOWATT[1], CHICAGO, ILL.

Member

A SURVEY of the heating and ventilating systems in use in the Chicago public schools will show examples of almost every type of heating and ventilation which has been in general use at any time throughout the United States. This is a result of the length of time the different plants have been in service and the ideas of the different designers during that time. In general, however, it may be stated that the Chicago practice in public school heating and ventilation is a plenum system with direct radiation only in offices, corridors, gymnasia, shops or other special rooms. The so-called "split system" of heating and ventilation wherein the heat losses in the class rooms are taken care of by direct radiation in the rooms themselves, with the incoming air warmed to a point where it is suitable for ventilation purposes only, has not been installed in buildings erected in Chicago for some years. It has been the opinion in Chicago that the class rooms are more certain to have ventilation when the heating of the rooms depends upon a predetermined air supply.

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[1] Chief Engineer, Board of Education, Chicago, Ill.

Paper presented at Semi-Annual Meeting of AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Chicago, Ill., May, 1923.

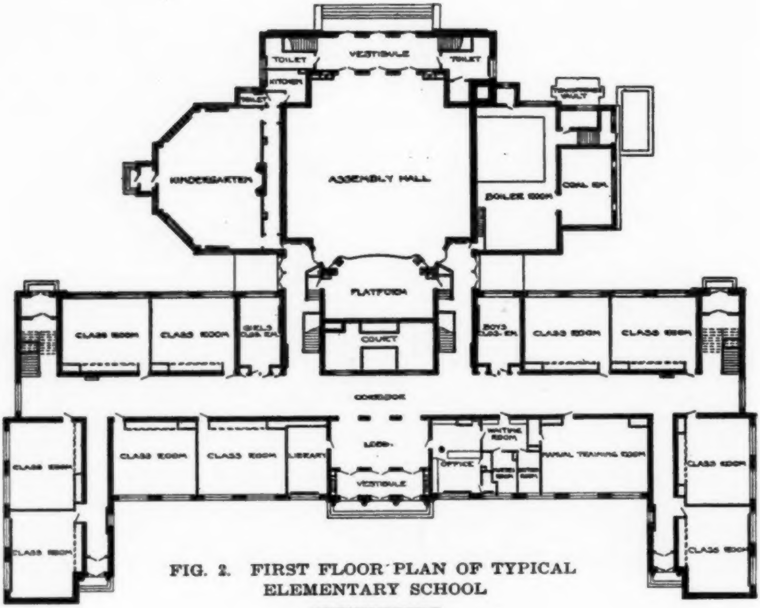

FIG. 2. FIRST FLOOR PLAN OF TYPICAL ELEMENTARY SCHOOL

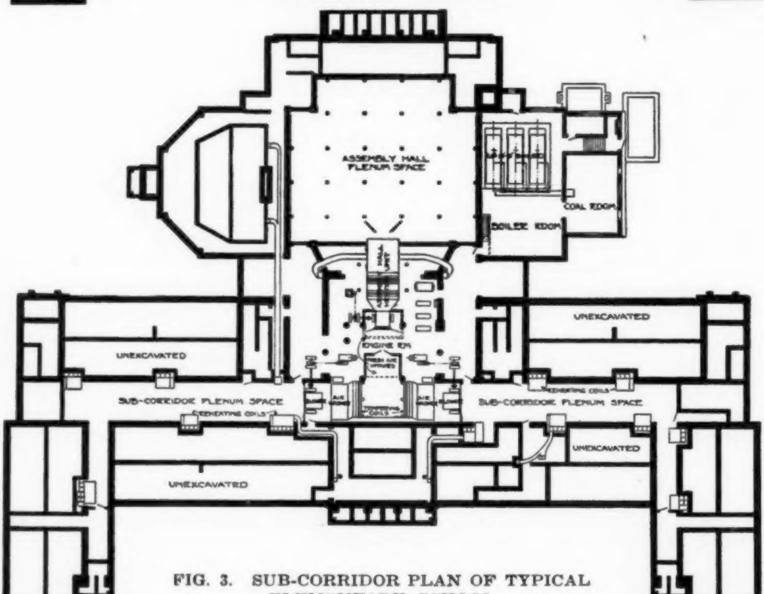

FIG. 3. SUB-CORRIDOR PLAN OF TYPICAL ELEMENTARY SCHOOL

NEED OF CAREFUL PLANNING

In the proper design and installation of any heating and ventilating plant a thorough co-operation between the architect and engineer must prevail in order to avoid having the architect proceed too far with the plans of a building before consideration is given to the proper design of the heating and ventilating system. Everyone has seen buildings constructed wherein it was evident that the architect in designing them had no thought whatever as to the mechanical equipment necessary to operate the building after it was completed; with the result that the boilers and mechanical equipment had to be installed in whatever space was left after all other demands were satis-

FIG. 4. CRANE TECHNICAL HIGH SCHOOL BUILDING

fied. It therefore not infrequently happens that, boilers, for example, have to be fitted into cramped, out-of-the-way places, so congested that the firemen are scorched in in their attempts to clean out the fires, jointed flues and tubes cannot be removed if they fail by rupture or corrosion; if those who design and install plants in this way were required to operate them the design and installation would be different.

THE CHICAGO PLAN

The situation in the Chicago public school system is very favorable because the Board of Education engineering and architectural staffs work together on all plans, co-operating in every way, and because those who are responsible for the design and installation of the plants are also responsible for their maintenance and operation after installation. This avoids all division of authority and makes for designs that are economical and practical.

The operating engineering force in the Chicago public school system is a well organized force of men who are represented by com-

mittees; these men are invited into the office of the chief engineer to go over proposed plans of new buildings and additions with a view of getting the suggestions of these practical operating engineers as to the workability of the proposed designs. It is our belief that in order to induce men to give proper care and attention to mechanical equipment, that equipment must be made readily accessible and proper working space must be provided. The design of the plants in the

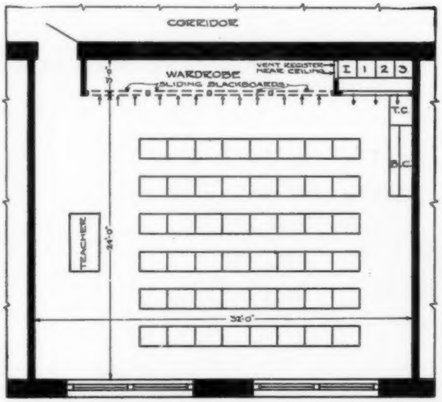

FIG. 5. PLAN OF TYPICAL CLASSROOM

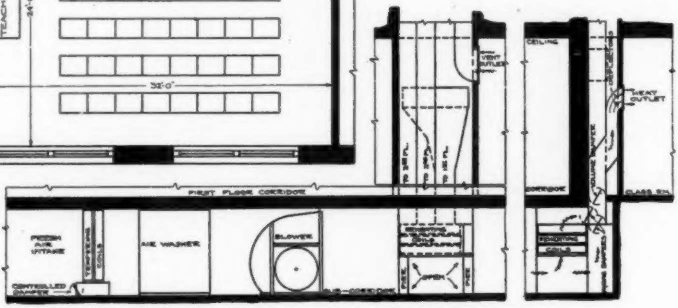

FIG. 7. TYPICAL SECTIONS THROUGH CORRIDORS

Chicago public schools is such as to give ample working space in all boiler, engine and pump rooms.

BOILER PLANT DESIGN

There are over 800 high-pressure boilers in use in the Chicago public schools at this time. About 100 of these boilers are water-tube boilers and most of the remainder are standard return tubular boilers. A typical Chicago elementary school consists of a three-story building containing 32 class rooms, assembly hall, gymnasium, offices, etc. (Taylor School, Fig. 1). This size of elementary school building is duplicated oftener than any other. The typical boiler room equipment for such a building consists of three 66 in. by 18 ft. standard horizontal-return-tubular boilers with shaking grates for hand firing. For plants of this size it has been found that this boiler is easy to keep clean and easy to repair.

The boilers are set with what is known as the Chicago Smoke-Department No. 8 furnace setting, consisting of brick bridge walls,

wing wall piers and deflecting piers. This setting is efficient and the upkeep cost is low, but has the objection that with this setting it is difficult to eliminate smoke. This city is located in a territory where all of the larger plants burn the lower grades of bituminous coal mined in Indiana and Illinois. If constant efforts are not made to minimize smoke, the city would be covered with a cloud of filthy soot. School plants are especially difficult to operate without excessive smoking, because boiler settings are cold when the plants are started up in the morning, and because the load tapers off so materially after the first few morning hours, giving an excess boiler capacity with corresponding low furnace temperatures. All of the boilers in Chicago public schools are equipped with high-pressure steam jets over the

FIG. 6. SUB-CORRIDOR OF TAYLOR ELEMENTARY SCHOOL BUILDING

fire doors which are very useful in mixing the gases of combustion before they pass over the bridge walls and assist in cleaning smoky stacks. Experience here, however, has shown that the best furnace for use in school work from the standpoint of smoke prevention is the dutch oven, or full-extended-type furnace.

When the boiler load is in excess of 400 hp., it is the practice in Chicago to install water-tube boilers, instead of the standard return-tubular boilers. These water-tube boilers are of the longitudinal-drum type with horizontal baffles. If the load is large enough to warrant it, it is the practice to install automatic stokers in connection with these boilers. The stokers are of different types, there being in use in the schools, chain grate stokers, Cox stokers, Model stokers and Swift automatic rotary stokers.

In the two high schools now under construction, the Crane and Tilden High Schools, the boiler plant of one is equipped with Burke natural-draft chain-grate stokers and the other with Harrington forced-draft chain-grate stokers. With the larger plants equipped with stokers, coal-handling apparatus is provided to facilitate distribu-

ting and storing the fuel. These conveyors are either of the endless-bucket Link-Belt conveyor type or a combination of bucket and worm.

For a number of years the school buildings in Chicago have been built three stories in height without basements. It is the present practice, however, to drop the coal-room and boiler-room floor levels as far below grade as the sewer-drainage will permit for the purpose of obtaining a larger available coal-storage space. The most usable space in coal bins is the space below grade. A deep coal bin is more satisfactory than a larger bin with the floor near grade level, but makes necessary some form of ash handling machinery. Ash handling equipment used in the Chicago public schools is of two types; the endless-chain-bucket type and the steam-ejector type. Because of the low cost of installation and the ease with which ashes can be delivered to any desired location, the steam-ejector type of ash-handling equipment is installed more frequently than is the bucket type. The ash room is usually provided by constructing a reinforced concrete floor across the upper part of the coal room with ash chutes to deliver ashes into trucks on the driveway.

In considering whether or not apparatus intended to increase the efficiency of any plant shall be installed it must be borne in mind that the school boiler plants have a large excess capacity except in the very severest weather. Heating plants in Chicago schools are designed to take care of the heating load with outside weather conditions 10 deg. below zero. This condition has to be met only a few times each year; moreover school plants are operated in this climate only about 150 days out of 365 and on some of these days only a few hours. Equipment designed to increase the efficiency of the plant and thereby reduce operating costs, and which would be a good investment in a commercial plant that runs nearly at full capacity and a much larger percentage of the time than do school plants, might be a very poor investment in the school plant. For that reason the equipment in the plants in Chicago schools has been kept simple and plain, as the possible saving effected by the additional equipment would not return its cost during its life of service. This is one reason why the installation of coal-handling apparatus and stokers has been confined in Chicago to the larger high school plants only.

PUPILS HEALTH PARAMOUNT

Every consideration must be given to safeguard the health and comfort of young people in a community while in the public school buildings. The attendance at a school in this country is compulsory. Boys and girls of school age are compelled to go to school and their health while doing so must be given consideration even above proper teaching methods. It is difficult to offset the damage that may be done by the spread of infection or contagion in a school building and is inexcusable if brought about by an unsanitary condition of any part of the building, including the air in the building. Uncleanable

air ducts or flues in any part of a ventilating system should be generally recognized as an unsanitary condition in an important part of the ventilating system.

For many years new school buildings erected in Chicago have been constructed without basements and a typical plenum system of heating and ventilation was used wherein ducts extended horizontally from a central plenum chamber along between corridor walls frequently more than 100 ft., then up vertical flues to the class rooms. For the sake of appearance these ducts in finished corridors were covered by a false ceiling.

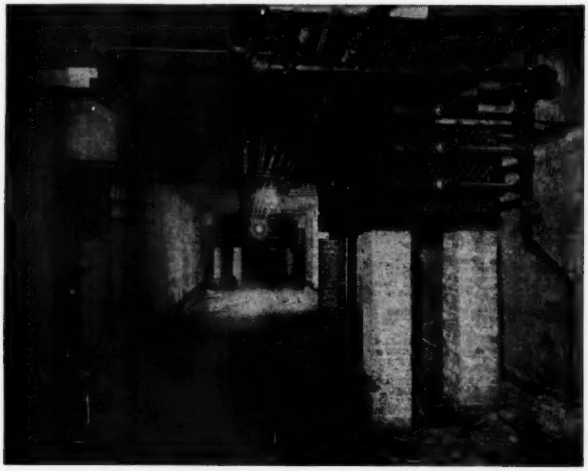

FIG. 8. CRANE TECHNICAL HIGH SCHOOL BUILDING SHOWING PIPE CONNECTIONS IN SUB-CORRIDOR

The Chicago Commission on Ventilation made a study of school ventilation among other problems and raised an objection to these long, inaccessible horizontal ducts on the grounds that they could not be cleaned out, were unsanitary, and were in part at least to blame for the peculiar, stale school-house odor sometimes noticed. It is a well known fact that dust settles on horizontal ledges and flat surfaces, and that the bottom inside of the horizontal air ducts soon became coated with a layer of dust and dirt that may be a breeding place for bacteria. This condition was forcibly drawn to the writer's attention some time ago by an incident that occurred at one of our public schools: The governor belt on the engine driving a steel-plate blower in one of the older buildings broke during school sessions, permitting the engine to race, driving the blower at high speed. The increase in the air velocity through the ducts stirred up and moved much of the dust that had been lying in the bottom and carried it into the rooms

in such clouds that the principal thought the building was on fire and dismissed the pupils by sounding a fire alarm.

VERTICAL SUPPLY DUCTS

The desirability of eliminating these inaccessible horizontal ducts brought about the development of the sub-corridor system to suit the needs of Chicago conditions and this arrangement was tried in two buildings about four years ago. It proved so satisfactory that the Chicago Board of Education directed that all new buildings be provided with ventilation systems of this type and this is the system being installed in all of our new buildings today. The Chicago experience with 10 new buildings operating with nothing but vertical supply ducts has proven such an improvement that any system that employs horizontal inaccessible ducts in any part of the air supply system cannot be considered perfect.

In the system adopted in Chicago the space between the corridor walls below the first floor is excavated to a depth of about 8 ft. These corridor walls are recessed back under every tier of rooms to provide space for three vertical air supply flues, one to the first-floor class room, one to the second-floor class room, and one to the third-floor class room. The walls, floor and roof of this sub-corridor space is of smooth finished concrete and is about 7 ft. high in the clear from floor to ceiling and the full width of the corridor, or about 14 ft. in width. Blowers draw air from fresh shafts through tempering coils and air washers and deliver it at a temperature of 65 deg. and with a relative humidity of about 40 per cent into these sub-corridors, which are thus used as tempered-air plenum chambers. Vento heating stacks are located at the recesses in the sub-corridor wall below the vertical air flues to the class rooms. The usual arrangement of mixing dampers, controlled by room thermostats, permit any mixture of tempered or heated air necessary to maintain the proper room temperatures. These sub-corridors are electric lighted and equipped with water connections and drains so that they may be hosed out, and are large and high enough so that those operating the plants can get about in comfort.

Blowers are selected which will deliver the computed quantities of air and maintain a static pressure of ½ in. of water in the sub-corridor space, and adjustable volume dampers are set in every duct. Because of the high static pressure carried in this large tempered-air plenum chamber, opening a class room window will not unbalance the ventilating system to such an extent as to interfere materially with the ventilation of other class rooms, the disturbance being confined to the room in which the windows are opened. Those who have to deal with 10,000 or more teachers and principals will know the advantage in having the individuals free of the feeling that they are required to work in a sealed-up space and must not under penalty open a window.

The system described provides no place for dust or dirt to settle; air is delivered into the class rooms just as it leaves the air washer;

the system is very flexible, making each group of three class rooms practically a separate unit; is adjustable to meet wind and weather conditions; adjustable for a partial use of the building and the resistance through the duct system being so very low it is found a considerable air circulation is obtained into the rooms even when fans are not in operation.

CLASS ROOM EXHAUST SYSTEM

The plan used in exhausting the air from the class rooms in the elementary schools in Chicago differs slightly from that most generally used. Cloak rooms are no longer provided. In lieu of cloak rooms, a wardrobe space about 30 in. in depth is provided along the corridor wall of each class room. In front of this wardrobe space, sliding hyloplate blackboards are provided which are counter balanced so they may be raised and lowered like a window sash, and when lowered, which is the normal position, drop to within 4 in. of the floor. A vent register is placed in the upper part of each wardrobe space and a duct extended to the attic and there connected to the main ventilators. The exhaust air is thus exhausted from the class rooms through a slot at the floor 4 in. in width, extending along about one-half the length of the room. This arrangement is sanitary, makes for a good distribution of air in the room, and carries all odors from coats and other clothing hung in the wardrobe directly out to the attic vents.

An accompanying plan shows a typical class room arrangement used in Chicago. The location of the windows will determine the direction which the pupils' desks will be faced and the location of the teacher's table. Chicago class room design provides only one door from each class room. As a matter of control that door should be located at the same end of the class room as the teacher's desk. Wherever possible the air inlet to the class room is located at the opposite end of the wall from the door and the incoming air is directed across the room against the windows and about 8½ ft. above the floor line. This is found to be the proper location for this air inlet to avoid short circuiting of the air out through the class room door whenever it is open, and to avoid drafts on the teacher's head.

CALCULATING HEATING AND VENTILATION REQUIREMENTS

In the calculation of requirements for the heating and ventilation systems of the Chicago public schools, the quantities of air required for class rooms, lecture rooms and study rooms are based upon the traditional supply of 1800 cu. ft. per hr. per seat; assembly halls 1500 cu. ft., and other rooms such as offices, dining rooms and libraries are calculated for a 10 min. air change; corridors a 20 min. air change; gymnasia, because of their large volume compared with the number of occupants, a 20 min. air change, and laboratories an 8 min. air change. The total air quantities thus found are used in a rough determination of the pounds of steam which will be condensed in

heating the air from an outdoor temperature of minus 10 deg. to a final plenum-chamber temperature of 130 deg., it having been found by experience that with the construction and design of school buildings such as is practiced in Chicago, the quantities of air named will furnish sufficient heat to the rooms to maintain a temperature of 70 deg. indoors with an outdoor temperature of minus 10 deg., when the final maximum temperature beyond the reheaters is 130 deg. Repeated

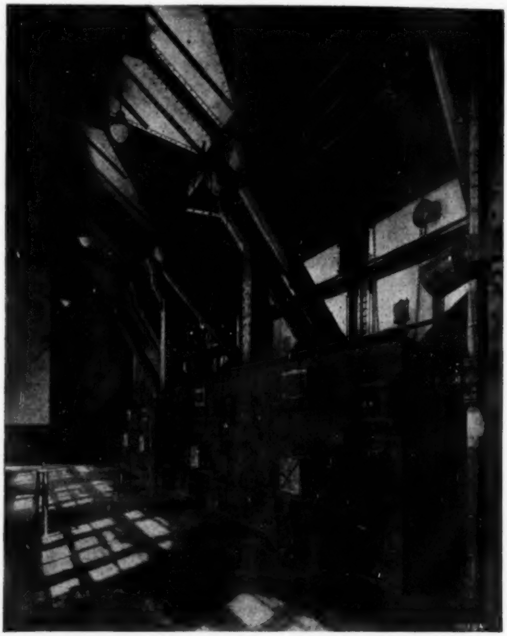

FIG. 9. STOKERS AND BOILER ROOM OF ENGLEWOOD HIGH SCHOOL BUILDING

heat-loss check and actual practice shows that this temperature will care for the heating requirements of an ordinary class room.

The assembly hall is usually ventilated by a separate unit consisting of a motor driven fan and draw through heater proportioned for a final temperature of 100 deg., the air entering through mushroom ventilators under the seats. Assembly halls are usually centrally located in the building and do not have the same proportion of exposure as the class rooms, so the final plenum chamber temperature does not have to be as high as the plenum chamber temperature for class room work.

Rooms that have an unusual exposure are provided with supplemental direct radiation. Rooms that it is expected will be used after

the regular school hours are supplied with direct radiation for heating after the fan engines are shut down. Direct radiation is also provided in vestibules and first floor corridors, and in other locations where cold outside drafts may enter. Air washers are used on class-room fans wherever the building is located in a district in which smoke or dust may be a nuisance and in every instance the new designs provide space so that air washers may be installed later if found necessary. Every building is equipped with humidifying apparatus consisting of steam-jet humidifiers controlled by automatically operated humidstats. The usual tempering heater units consist of Ventos figured for an air velocity of 1000 ft. per min. through the stacks. In the design of the sub-corridor systems reheaters are arranged for an air velocity through them of 600 ft. per min. The boiler plants are operated at about 100 lb. gauge steam pressure, but engines driving the blowers are designed to operate the blowers at full speed at any steam pressure between 50 and 100 lb. with 5 lb. back pressure on the engines. The engines exhaust through oil separators into the heat mains, the additional steam required for heating the building being supplied from high-pressure lines through reducing valves.

In general each fan chamber is treated as a unit, having its individual steam-supply lines from the boiler, return lines to the pump, pressure reducing valve and vapor lines to the roof. The room temperatures are maintained constant by automatic thermostats located in the rooms acting on double mixing dampers at the plenum chamber. Tempered air is automatically controlled by by-pass dampers located under the tempering stacks. No automatically controlled steam valves are used except on the humidifiers which are controlled by humidstats located in some typical room. The rules of the Board require class rooms to be kept at temperatures between 67 and 70 deg. with a relative humidity of 40 per cent, and apparatus is adjusted to control air conditions within this range. This temperature and humidity were established from a consideration of comfort rather than a consideration of health.

When the total steam requirements are known from the above calculations, allowing two air changes per hour for leakage and after adjustment to unusual conditions that may exist in different parts of the building, the boiler capacity is found by estimating 4 lb. of steam evaporated per square foot of boiler heating surface per hour. This provides a rough check as to the number of square feet of heating surface required in the boiler plant, and boilers selected accordingly. Whenever a building is designed which is radically different from others, or a room has an unusual location of exposure, the heat loss calculations are carefully made in accordance with the usual system of heat loss calculation methods to determine whether the air required for ventilation will meet the requirements for heating. It has been found, however, that theoretical calculations of heat losses, while useful in the preliminary studies must be checked against practical results our extensive experience has taught us to expect.

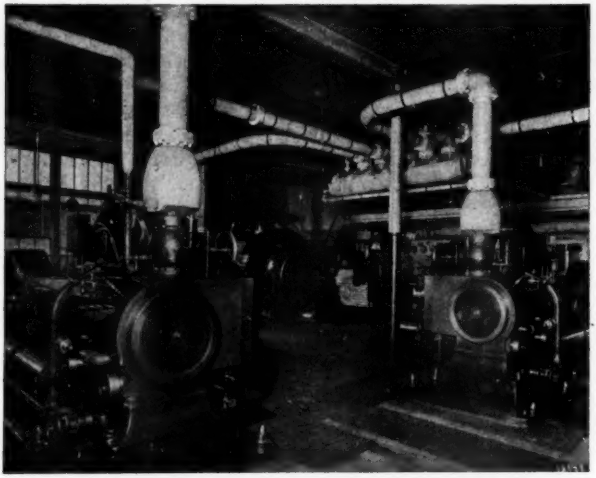

FIG. 10. LANE TECHNICAL HIGH SCHOOL BUILDING ENGINE ROOM

THE DRAFT PROBLEM

One of the most aggravating features with which those in charge of the operation of plants have to contend is the question of draft. An ample draft properly controlled is the best fuel economizer that

FIG. 11. PUMP AND TANK ROOM, TAYLOR ELEMENTARY SCHOOL BUILDING

can be provided. This feature was not given proper consideration in past years in Chicago with the result that the draft in many of the older buildings is quite inadequate. Of late years, however, we have been fortunate in having had the co-operation of the Smoke Department in this matter, so we now have established a minimum stack height from the boiler room floor to the top of 115 ft. It is found that this provides ample draft for an ordinary horizontal-return-tubular boiler plant and with the proper damper control will make it possible to regulate the combustion of almost any grade of fuel that must be burned in this district.

PLANT OPERATION COST

The cost of plant operation has received the attention of trustees since the war, because it has become so large an item in the annual budget. It requires approximately 150,000 tons of coal to heat and ventilate the school buildings in Chicago each year. The maintained high price of fuel makes it of very great importance that economies in its consumption be practiced wherever they can be practiced without injury to the school system. Twelve million cubic feet of air is drawn from outdoors through the fan systems in the school buildings every minute, is heated, and delivered into the rooms maintaining the temperature at about 70 deg. The average outdoor temperature in Chicago during the heating season is approximately 35 deg. When all of the air delivered into the rooms is exhausted through the vent flues to the roof, all of the fuel required to heat the air from outdoor temperature to 70 deg. is wasted, which in a climate such as is found in Chicago amounts to considerable.

Up to this time the Board of Education has in general been opposed to re-circulating the air from the class rooms, believing that it is safer from a health standpoint to draw in fresh air, heat and deliver it to the rooms and discharge it directly out of doors. The Chicago Board believes in taking no chances where the health of the pupils and teachers are concerned.

In modern school buildings, however, the assembly halls are made so large that there is considerable waste of heat which could safely be conserved. For example, the ordinary modern elementary school building has an assembly hall designed to seat approximately 1000 people and air has to be delivered into this hall to provide ventilation for that many seats. The hall is used to capacity only two or three times per year and this 1000 seat assembly hall is being occupied daily by perhaps not to exceed 50 people. However, because the hall is used by some classes sometime each day and because zone ventilation is not practical it is necessary to keep the hall heated and ventilated throughout the entire school session. Under such conditions air enters the vent ducts in practically as pure a condition as it was when it was delivered into the hall and is of such purity it may be used in the general air supply to the class rooms. The heating and

ventilating design of the new school buildings, therefore, provides for the re-circulation of air from the assembly halls when conditions are such it can be done without jeopardizing proper ventilation. The entire problem of re-circulation of air is being considered by a Committee on Health and Sanitation of the Chicago Board of Education, and if this Committee finds that general re-circulation may be practiced under proper conditions it will result in permitting the installation of smaller boiler plants and will result in a material reduction in operating costs. A study of the operating cost given in Tables 1 and

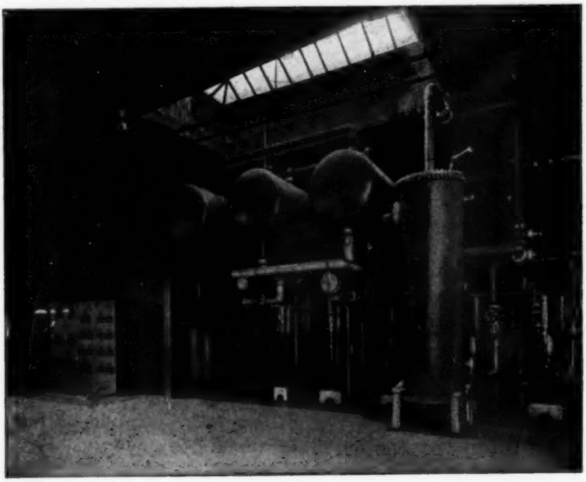

FIG. 12. PUMP AND TANK ROOM, ENGLEWOOD HIGH SCHOOL BUILDING

2, will show the importance of adopting every proper means of reducing them. In considering these tables of costs, it must be borne in mind that as Chicago schools are cared for on the so-called indirect employment system, the item *salaries* includes the salaries of the engineering and janitorial force, and the item *supplies* includes both engineering and janitor supplies.

TABLE I. APPROXIMATE COST PER ANNUM PER PUPIL—ELEMENTARY DAY SCHOOLS

Year	Fuel	Operation Supplies	Operation Salaries	Total Operation	Total Educational Including Operation
1917	$2.05	$.17	$3.97	$6.19	$44.30
1918	2.70	.17	4.30	7.17	47.95
1919	1.96	.19	4.37	6.52	53.48
1920	3.20	.21	5.20	8.61	62.87
1921	2.80	.22	6.21	9.23	70.37
1922	2.45	.22	6.17	8.84	71.51

TABLE 2. APPROXIMATE COST PER 100 CU. FT. OF HEATED AND VENTILATED SPACE—ELEMENTARY DAY SCHOOLS

Year	Fuel	Operation Supplies	Operation Salaries
1917	$2.41	$.20	$4.66
1918	3.18	.20	5.07
1919	2.22	.21	5.04
1920	3.83	.25	6.24
1921	3.47	.27	7.72
1922	3.07	.28	7.73

DISCUSSION

C. D. ALLAN: I would like to ask if siphon ventilators or something of that kind are used to assist the gravity exhaust, or whether it opens through a goose neck turned down.

JOHN HOWATT: There is a type of ventilating design which has some siphonic action, but we don't depend very much upon that plan to remove the air.

E. S. HALLETT: Mr. Howatt's paper certainly has described a very efficient ventilation system and I am delighted to know that he is moving so much air through his schools and that he has eliminated radiators from the rooms to the extent that he has.

One practice we vary on is that Mr. Howatt separates his Vento coils in ducts, while my policy is to place them in a concentrated mass and move all the air, after it has been heated and washed, etc., through this duct. My opinion is that dry cleaning for a mass of coils like that mentioned is far superior to wet cleaning, for when water is used mud results, and if the janitors do not remove it carefully, dust will form, and dust, without question, is the greatest difficulty we have to contend with in our work.

On the matter of re-circulation, the results of our six or seven years' work when stated astonish everybody. In the latest St. Louis school building we are re-circulating 90 per cent of the air with the fresh air intake wide open all the time. Why this can be done will be obvious when the method is studied. Though we are a little bit farther south than Chicago, in this best school of ours the heating cost was 56 cents per pupil against Mr. Howatt's $2.15 during the past year. The entire per capita cost for fuel last year was $1:14 in St. Louis and I understand Kansas City reports $3.04; Chicago's is $2.15.

C. W. MILLER: What type of drain is used to carry away the waste water after these corridors have been washed down? Also have the seals ever been broken by the air travel or evaporation of that water?

JOHN HOWATT: The drain is through a deep seal trap outside the corridor; the door sill is cut; the entrance to this corridor chamber is cut and the trap is outside to avoid the danger of drying out in the trap.

P. J. DOUGHERTY: I should like to emphasize a statement or two in this excellent paper, first, that on the draft problem, which says, "An ample draft properly controlled is the best fuel economizer that can be provided," and also that stacks not less than 115 ft. high are used.

J. G. FINAN: I should like to ask where this mud comes from through the air washer, as mentioned by Mr. Hallett.

E. S. HALLETT: Some dust must get through the washer and the coils and when these are wet, mud forms. Now I am sure with an air washer such as Mr. Howatt has in his schools and we have in all our schools that we built many years back we have a minimum of that. but no air washer ever built takes out more than 80 or 90 per cent of the dust. There is microscopic dust that gets through and when it gets wet it makes mud.

J. G. FINAN: Do you think if you hose it out twice a week it is better to use a vacuum cleaner rather than water to wash that out through an open draft?

E. S. HALLETT: I think there is no other way in the world as good as the vacuum cleaner for removing dust. I am sure it is cleaner. I would be willing to go on record to have a test made on that, i. e., that a vacuum cleaner would remove a larger portion of injurious dust or dirt that would accumulate than wet cleaning would, unless very great quantity of water is used.

No. 669

A FIELD METHOD FOR DETERMINATION OF OZONE

BY W. P. YANT[1], G. W. JONES[2] (Non-Members)

AND

F. C. HOUGHTEN[3] (Member)

PITTSBURGH, PA.

INTRODUCTION

AT a previous meeting of your Society[4] the authors reported the results of tests made by the Bureau of Mines in co-operation with the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS on the amounts of oxides of nitrogen produced by two types of ozone apparatus used for ventilation purposes. Tests have now been completed on four types of ozone apparatus, three for ventilation and one for industrial purposes. From these tests it can be safely said that dangers from oxides of nitrogen, from a physiological standpoint are exceedingly remote with the apparatus designed for ventilation. In no case were oxides of nitrogen found in proportions greater than 1 part to 25 of ozone. In circulated air treated with ozone the proportions would not be greater than 1 part in 50 million assuming that the ozone concentration does not exceed ½ part per million which due to its penetrating and disagreeable odor seems to be the upper limit that may be used. Physiologists report no ill effects from oxides of nitrogen[5] even when present in concentrations of 1 part per million. It is thus seen that fifty times the above amount of oxides of nitrogen could be present without causing physiological harm.

Having found that oxides of nitrogen occur in only very small proportions, attention was directed to devising a field method for the determination of ozone at concentrations of 2 parts per million or less, whereby it

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[4] Annual meeting AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, New York, January, 1922, and published, Jones G. W., Yant, W. P., and Armspach, O. W., Oxides of Nitrogen Produced by Ozone Apparatus; JOURNAL A.S.H.-V.E., Vol. 28, p. 191.

Paper presented at the Semi-Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Chicago, Ill., May, 1923. Published by permission of the Director Bureau of Mines, Department of the Interior.

is possible to accurately establish and safely control the use of a non-irritating concentration.

When dealing with a problem so vital as that of modifying the air breathed by man it is indeed essential that we proceed with caution, especially in prescribing the addition of a gas-like ozone, which is quite irritating to the lung tissues when present in even relatively small amounts. However, it is not unusual for gases to be irritating when present in concentrations higher than those ordinarily experienced for it has been observed that oxygen, even though it is essential to life, will, when present in four to five times the amount found in normal air cause death to animals through lung irritation resulting from three to four days exposure. Nature controls the concentration of oxygen in normal air to that to which the lungs are adapted and this must likewise be done if it is desired to modify the composition. Thus it is necessary that methods of control be designed which will fulfill the requirements.

The method herein described is made as simple as possible, with robust apparatus not easily broken, so that its use need not be limited to trained analytical chemists. The chemicals used remain fairly constant over extended periods of time, and the method is such that tests can be made in school rooms with a minimum amount of disturbance to the occupants.

PRINCIPLES OF METHOD

The method depends on the familiar reaction of ozone with potassium iodide.

$$2\ KI + O_3 + H_2O = 2\ KOH + O_2 + I_2$$

and a subsequent determination of the iodine using starch indicator as a measure of the ozone. Numerous methods involving this reaction have been developed but they have been open to criticism, not on the validity of the reaction being quantitative but on the manner in which the analysis is conducted, these criticisms being primarily as follows:

1. In passing a sample of gas through an absorption bottle containing the *KI*, some of the liberated iodine is carried off as vapor by the effluent gas, thus giving results that are too low.
2. In the time required for passing a large sample of gas through an absorption bottle at a rate slow enough to permit sufficient contact between gas and liquid for complete absorption, there may be a source of error in the liberation of iodine by the action of light, heat, or other effects, for it is known that a solution of *KI* will decompose slowly on standing.
3. Thiosulphate solution of a strength suitable to determine the small amount of iodine liberated, deteriorates very rapidly.
4. The blank on the reagents may far exceed the equivalent of the gas present.
5. In the ordinary titration of iodine, until the blue color of starch is entirely dissipated, ordinary differences in deciding the end point may be greater than the equivalent of the gas present.

In the following method these objectionable features are practically eliminated through the manner in which the sample is brought in contact

[8] Loc. cit.

with the gas; the compensation for light, decomposition, and blank effects, and the indirect method by which the iodine is determined.

DESCRIPTION OF METHOD

The method is essentially an adaptation of the apparatus and procedure used by the Selby Smelter Commission[6] in their extensive investigation of small amounts of sulphur dioxide in atmospheric air. In brief this method consists of obtaining a sample of the air in a partially evacuated 24 L

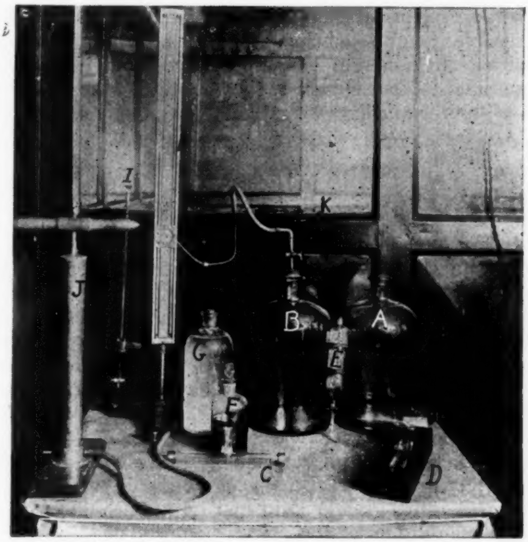

FIG. 1. VIEW OF FIELD APPARATUS FOR DETECTION OF OZONE

sample bottle and absorbing the SO_2 by shaking with a solution of starch and KI made slightly blue with free iodine. The resulting color is compared in test bottles to an equal portion of absorbing solution from the same stock similarly shaken in another sample bottle with pure air. Any loss of color due to the SO_2 removing some of the iodine is restored by the addition of standard iodine from a burette. From the volume of iodine solution required the volume SO_2 is calculated.

The reaction in the determination of SO_2

$$SO_2 + I_2 + 2H_2O = 2HI + H_2SO_4$$

in which free iodine goes into combination and a decrease of color occurs, is opposite to the one with ozone in which iodine is liberated from compounds of iodine, thus increasing the amount of color. However, the

[6] Report of the Selby Smelter Commission, by J. A. Holmes, E. C. Franklin, and R. A. Gould; Bureau of Mines Bulletin 98.

only difference in the determination is that rather than having to restore the blue color to equal that of the control as in the case of SO_2, the color of the control is made to equal that resulting from the sample containing ozone. Otherwise the procedure and calculation are similar.

LIMITS OF THE METHOD

The method as herein adopted for the determination of ozone is designed to cover the range of concentration that would be of practical use in ventilation work. It has been observed by Hill[7] and also the authors that even with 1 p.p.m. present the odor of ozone is so penetrating as to be disagreeable if not deleterious, and on this basis it is only necessary that the method cover a range having a maximum of 2 p.p.m., thereby making it possible to concentrate the accuracy in that region rather than providing for greater but impractical amounts at the expense of accuracy.

APPARATUS AND SOLUTIONS

The apparatus as first used by the writers was similar in all respects to that described in detail in the above report[8]. But it was soon found that the outfit as designed by the Commission for their portable laboratory was too large and cumbersome to be otherwise transported from place to place. In view of this the size of the sample bottles was reduced from 24 to 10 liters. Also the amount of absorbing solutions was reduced in even greater proportion. The test bottles to which the starch solution was transferred from the gas sample bottles for comparison were changed in design from ordinary round bottles to long tubes. By this device the thickness of solution observed was for equal volumes greatly increased over that of plain bottles and consequently any changes in color would be proportionately greater. With the particular tubes used the thickness observed was 3½ times greater than that of the original test bottles, thus increasing the magnitude of the color change a like amount.

Since the observed color produced by one drop of standard iodine solution was likewise increased, a more dilute solution could be used which in turn also tended towards greater accuracy.

The apparatus finally adopted for the determination of ozone in air is shown in Fig. 1 and consists of the following:

A and *B*. Sample bottles. Two 10-liter aspirator bottles having 1-hole stoppers in the tubulures. The stopper in the top of the bottle to be used for obtaining ozone-air mixtures is fitted with a glass stop-cock, all the others being fitted with glass plugs.

C. Comparison tubes. These are of equal length and were made from ordinary ¾-in. glass tubing with clear glass windows sealed on both ends. The capacity was 125 c.c. when the upright stem was 1/3 filled. Each tube is fitted with a 1-hole rubber stopper having a glass plug to exclude the air except during titration.

E. Measuring funnel. A measuring funnel from which equal portions of the starch solution could be added to each of the sample bottles through the lower tubulure

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[7] Hill, E. V. and Aeberly, J. J., What About Ozone? *Heating and Ventilating Magazine*, Vol. 18, 1921, p. 29.

[8] Loc. cit.

was made by bending the stem of a 300 c.c. separatory funnel at right angles, thus facilitating the introduction of the solution when the sample bottles are in an upright position. 125 c.c. calibration marks on the funnel indicate the volume of liquid introduced.

J. Pump. The pump used was an ordinary laboratory double acting foot-pump as used for sampling gases.

I. Burette. The burette was of a special type calibrated to 0.02 c.c. marks making an estimation of 0.01 c.c. very easy.

D. Comparison box. This was made as a convenience for holding the comparison tubes during observation. The front end consists of a block having two holes of the same diameter as the tubes tangent to each other and into which the tubes are inserted during comparison. A piece of ground glass immediately in front of the windows serves to diffuse the light.

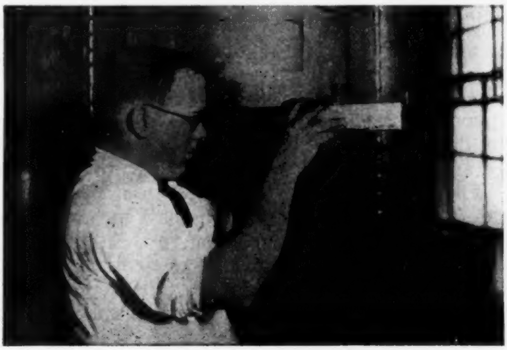

FIG. 2. ESTIMATING PERCENTAGE OF OZONE WITH COMPARATOR

H. Mercury gauge. Any open U gauge is suitable and need not be as elaborate or as long as the one shown in Fig. 1.

F. Iodine. 0.127 g. iodine per liter = N/1000; 1 c.c. N/1000 solution =o= 0.000024 g. O_3 = 0.0112 c.c. O_3 at 0 deg. and 760 m.m. Hg. This is made according to the procedure given in any standard text book on quantitative analysis and if carefully weighed need not be standardized for this work.

G. Starch solution. Nearly any kind of starch (potato, grain or soluble) is suitable. For most of the work potato starch was used. This is prepared in the usual manner by adding 1 gram of dry starch which had been mixed with sufficient cold water to make a thin paste, to 1 liter of boiling water, and after boiling for several minutes allow to cool and settle. A good practice is to make the starch the day before, for each succeeding day's work. The starch will settle over night and the clear solution above can be syphoned off, it being more satisfactory to work with than turbid solutions. Ten grams of *KI* is added to each liter of starch solution before using for absorbing the O_3. Also the solution is made slightly acid by the addition of two or three drops of concentrated sulphuric acid per liter.

PRELIMINARY PROCEDURE

Add iodine to the starch-*KI* solution to bring it to a color which is faintly but distinctly blue when in the comparison tubes. Fill the measuring funnel to the 250 c.c. mark and introduce 125 c.c. into each of the comparison tubes and compare the color by observing them as in Fig. 2,

to make sure there is no effect due to dirt or foreign matter. One hundred and twenty-five c.c. of the tinted solution is in a similar manner transferred to each of the gas sample bottles, replacing the plugs and closing the cock thereafter. The bottles are then shaken vigorously for 5 minutes to wash them free from oxidizable or reducing matter or gases, after which the solution is transferred to the comparison tubes, in so doing the plug in the lower stopper being replaced by a short piece of glass tubing. If the colors are not the same the operation should be repeated with another portion of the tinted solution and so on until the colors do not differ by more than 0.02 c.c. of N/1000 I_2. Usually, leaving some of the absorbing solution in the bottles each night will reduce the number of trials to one or two. When the two solutions have the same color the sample bottles are ready for analysis.

ANALYTICAL PROCEDURE

One hundred and twenty-five c.c. of fresh starch solution is added in the regular manner to each of the sample bottles. The plugs are replaced in the control bottle *A*, but *B* is connected by the glass stopcock to the mercury gage and by a short hose connection to the exhaust side of the foot-pump. The bottle is then evacuated to a degree which will be explained later, usually approximately one-half the prevailing barometric pressure, after which the hose connection to the pump is closed with a screw clamp and after observing the mercury column for a few seconds to detect any leaks, the stopcock is closed. The mercury gage and pump are disconnected and the bottle carried to the place of sampling. When the sampling place has been reached the stopcock is opened and bottle *B* allowed to resume atmospheric pressure, thus instantaneously obtaining a sample of gas.

Both bottles are then vigorously shaken 60 times, endeavoring to wet all the surface of the bottle and to get contact between the gas and the absorption liquid but avoid unnecessary washing it against the rubber stoppers. After shaking transfer the solution to the comparison tubes (removing and then replacing all plugs and stoppers) and note any difference in color. If ozone is present in quantities greater than 0.05 parts per million the tube representing bottle *B*, or the actual gas sample, will be distinctly darker and the iodine solution should be added (by removing the glass plug and inserting the stem of the burette well through the hole in the stopper) drop by drop until the colors are equal, making sure to thoroughly mix the contents of the comparison tube after each drop before making observations. This mixture can be done by inverting the tube once or twice, allowing the solution to entirely displace the air in the neck.

If no determinable color change occurs a second determination should be made as described before, but after shaking 60 times the bottle should be re-evacuated and another sample drawn in and the shaking repeated before color comparison is made. The control should likewise be shaken 120 times. This operation can be repeated three or four times if necessary, thus making it possible to get a definite measurable reading with but small amounts of ozone 0.02 p.p.m. or less.

Should the solution appear too dark for comparison as will be the case if more than 2 p.p.m. of ozone is present another 125 c.c. portion of the starch solution should be shaken in the control bottle and the original solution diluted until the comparison can be made. Calculation is then based on the volume of iodine used times the dilution factor. In making these dilutions undue exposure to air that might contain ozone should be avoided.

Another manner in which these higher concentrations may be determined is to have sets of comparison tubes of equal volumes but different lengths and diameters, or adjustable tubes which can be obtained from apparatus dealers.

Calculation.—The volume of ozone is calculated from the iodine equivalent of the solution used.

From the reaction, $2KI + O_3 + H_2O = 2KOH + O_2 + I_2$

1 c.c. of normal iodine solution = 11.2 c.c. O_3 at 0 deg. and 760 m.m. Hg or 12.6 c.c. at 25 deg. cent. and 740 m.m. Hg,

the latter for all practical purposes being considered as room temperature.

Then 1 c.c. $\frac{N}{1000}$ iodine = 0.0126 c.c. O_3 at 25 deg. cent. and 740 m.m. Hg, or 0.01 c.c. $\frac{N}{1000}$ iodine = 0.000126 c.c. O_3.

The volume of sample is calculated from the ratio of the vacuum produced, as determined by the mercury gage, to the prevailing barometric pressure, times the total volume of the sample bottle, or,

$$\left.\begin{matrix}\text{Volume of}\\ \text{gas sample}\end{matrix}\right\} = \frac{\text{Mercury gage differentia}}{\text{Barometric pressure}} \times \left\{\begin{matrix}\text{volume of}\\ \text{sample bottle}\end{matrix}\right.$$

$$\text{Parts per million} = \frac{12.7 \times \text{c.c. } I_2 \text{ used}}{\text{Volume sample in liters}}$$

ACCURACY OF METHOD

One-half drop of N/1000 iodine solution produces the limit of distinction in color change, this being approximately 0.02 c.c., is equivalent to approximately 0.04 p.p.m. of ozone in air, using a 5-6 liter sample. However, due to other causes, the writers have found that with the above sample volume the limit of accuracy is 0.1 p.p.m., there often being slight differences in the solutions in the comparison tubes due to other causes. By the re-evacuation process this sample can be increased two, three, or even four times, it being useless to extend it further, and from experience the amount of iodine liberated has been found to be accumulative with the error factors remaining practically what they were for one evacuation. This makes it possible to accurately determine small amounts which with one evacuation would be seriously affected by the ordinary errors incurred. The limit of accuracy can be stated approximately as follows:

For 5-6 L sample	....	0.10 p.p.m.
For 10-12 L sample	....	0.05 "
For 15-18 L sample	....	0.03 "
For 20-24 L sample	....	0.025 "

It should be repeated that the results in all cases will be more dependable if the liter is 0.10 or more c.c. of iodine solution. When less than this the work is too near the limiting factors of the method.

If some of the absorbing solution is left in the sample bottles and the openings into the bottles kept closed the preliminary conditioning procedure should not take longer than 10 to 15 min. After this has been done a single determination, using a sample obtained by one evacuation, can easily be made in three to five minutes. The rapidity with which the determinations can be made permits a rapid canvass of the concentrations existing in large school buildings and other similar buildings having a great number of rooms.

There may be contaminating gases, principally oxides of nitrogen and sulphur dioxide, present in quantities to affect the results, the former causing high results while the latter will cause them to be low. It has been found[9] in previous work as well as in a continuation of that work which has been included in this report that the oxides of nitrogen given off by the ozone generators is not sufficient to cause any serious error in the determination of ozone in this field of use. In order to ascertain the presence of these two interfering gases a duplicate sample should be slowly drawn into the sample bottle through an absorption tube connected to the glass stopcock of bottle *B* and containing a 3 per cent solution of sodium hydroxide, which will remove them, allowing the ozone to pass through without appreciable alteration.

PRECAUTIONS

As a guard against possible contamination by outside air the Selby Commission[10] advises that the absorbing liquids be transferred from the stock bottle to the gas sample bottles and these to the comparison tubes in as nearly as possible a closed system. The writers have deviated from this in transferring the absorption solution to the comparison tubes since the determinations will be made in an atmosphere of the same composition as the sample and if the work is done rapidly an appreciable contamination will not result. However, even this may be avoided by putting stoppers having two holes in the comparison tubes, one of them being fitted with a glass plug as before and the other a right angle glass bend which can be inserted directly with the lower opening of the bottles.

The control bottle and the actual sample bottle should be treated and exposed as nearly identical as possible, the action of direct sunlight and heat being avoided.

If the determinations are to be extended over an entire day, only half of the absorbing solution should be tinted at one time as it tends to darken on standing.

Where the sample bottles are allowed to stand over night or for any considerable period of time 200 c.c. of well colored absorption solution

should be contained therein. This will reduce time of the preliminary procedure for preparing the bottles for use.

In the construction of the apparatus, all rubber stoppers should be washed with dilute iodine solution to remove oxidizable matter.

The iodine solution should be kept in a glass stoppered bottle which has been painted black to exclude the action of light.

The color changes are more distinct if the observations are made against a clear, bright sky.

EXPERIMENTAL WORK

In order to verify certain phases of the procedure described, the following experiments were made. Only a few of the results obtained are given but they are typical and representative.

Experiment 1.—To determine whether or not 60 shakes would entirely remove the ozone. An ozone-air mixture was sampled and after 60 shakes the ozone was determined. 125 c.c. of fresh absorbing solution was then admitted to the bottle and the shaking repeated.

Results:

	Ozone p.p.m.
After 60 shakes	1.5
After 60 shakes with new absorbing solution	0.02

This test was repeatedly made after determinations in Experiments 2 and 3, but in all cases the amount of ozone remaining after 60 shakes was found to be negligible.

Experiment 2.—To determine possibility of duplicating results an ozone generator was kept running continuously in a room having a constant air change and several determinations made each day for five days, fresh starch being used for each day's work.

Results:

	Ozone determinations p.p.m. — 5 L Sample			Average	Maximum variation from Average
	1	2	3		
1st day	0.52	0.60	0.65	0.59	0.07
2nd day	0.63	0.68	0.70	0.67	0.04
3rd day	0.57	0.48	0.50	0.52	0.05
4th day	0.70	0.60	0.72	0.67	0.07
5th day	0.65	0.58	0.70	0.64	0.06

Experiment 3.—To determine possibility of re-evacuation process. Samples taken during fifth day of Experiment 1.

No. of evacuations	Volume of sample, liters	Ozone in p.p.m.
1	6.35	0.58
2	12.70	0.52
3	19.05	0.67
4	25.40	0.65

Experiment 4.—Efficiency of $NaOH$ scrubbing device for removing traces of SO_2 and oxides of nitrogen. After completion of Experiments 1 and 2 traces of SO_2 and oxides of nitrogen were added to the ozone-air mixture and analysis made on samples which had been slowly drawn through a 3 per cent $NaOH$ solution contained in an absorption bottle which was connected directly to the cock of bottle *B* after the latter had been evacuated.

[9,10] Work cited.

Results:

		N/1000 iodine titration	Equivalent of ozone p.p.m
1.	Before adding SO_2 and oxides of nitrogen but without scrubber	0.30	0.60
2.	Same as No. 1 but using scrubber	0.26	0.52
3.	After adding SO_2 and oxides of nitrogen but without scrubber	0.80	1.60
4.	Same as No. 3 but using scrubber	0.32	0.64

As a result of the experimental work the method described is particularly well adapted to the determinations of small amounts of ozone in air, such as may occur in ventilation work. The apparatus is rugged and easy to operate. The solutions are available from any chemical laboratory or dealer and require no great skill in preparation. Determinations can be made very quickly and results far in excess to the necessary accuracy can be obtained. In fact, the writers know of no other method even for laboratory use whereby an equal accuracy can be obtained so easily and quickly. And finally technique involves no operations that cannot be carried out by even a non-technical observer.

DISCUSSION

E. S. HALLETT: The findings of Mr. Yant in his very exhaustive investigation are in agreement with the results obtained by us in St. Louis. We have standardized upon an ozone concentration at the point just under the noticeable odor of ozone in ventilation work. The two determining points in the use of ozone that have to be settled before it could come into general use are, first, a desirable concentration and secondly to correct prevalent impressions about ozone.

DR. E. V. HILL: I am glad to see ozone being placed properly before the heating and ventilating engineering profession and these studies I think will show it as a useful adjunct in ventilation practice. While manufacturers of ozone equipment have advised using one part per million in school ventilation we have found that but one-tenth part per million was found to give a more comfortable and agreeable condition, so I am glad to hear Mr. Yant say that one-half part per million is about the maximum for comfort.

J. M. ROBB: It seems to me that the amount of ozone necessary to get a given effect is similar to the problem in heating where you must estimate the amount of heat loss in advance and put in enough capacity to replace this loss. I would like to point out that good sized quantities of ozone are produced by X-ray machines and most of them are in unventilated enclosures where the operators work for long periods but I know of no record of anyone being injured in this work.

W. P. YANT: I do not think that ozone concentration can be accurately determined by the sense of smell because this differs too much from one man to another and also from the atmospheric conditions both preceding and after the test. I would like to see Mr. Hallett check up his work by some chemical means. This article was prepared for the use of men working with ozone and if Mr. Hallett could get an accurate determined basis of the ozone he is using on large groups of children he could obtain some very valuable data.

No. 670

VENTILATION OF GARAGES

By G. W. Jones[1], and S. H. Katz[2], Pittsburgh, Pa.

Non-Members

AUTOMOBILES with gasoline driven motors discharge large amounts of poisonous carbon monoxide gas in the exhaust. Reports of deaths, due to carbon monoxide generated by automobile engines, run in tightly closed, unventilated garages, are not infrequent news items; such accidents occur mostly in small, single car garages; however, carbon monoxide also creates hazards in commercial garages and automobile shops. Of course when automobiles or engines are tested on a block or otherwise stationary, the exhaust gas should be discharged outdoors through pipes as is done in the large manufacturing plants, but this may not be possible when the cars move.

The Bureau of Mines has received inquiries from large garage operators regarding the feasibility of employing carbon monoxide gas masks to protect men in their employ, elevator operators and others, who may suffer headache or nausea from exhaust gas. The use of gas masks for such purposes has not been recommended; instead ventilation is advised, sufficient to dilute the exhaust gases below concentrations which may cause ill effects.

Two years ago 101 automobiles of various types were tested by the Bureau under different loads and operating conditions to secure data on ventilation requirements for vehicular tunnels now under construction between New York and Jersey City[3] and at Pittsburgh. Some of this data may be useful to ventilating engineers who may be called upon to provide equipment to remove automobile exhaust gas from buildings or other enclosed places; it is presented here in brief form with statements of

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[1] Asst. Physical Chemist, U. S. Bureau of Mines.
[2] Assoc. Physical Organic Chemist, U. S. Bureau of Mines.
[3] Fieldner, A. C., Straub, A. A., and Jones, G. W., Summary Report on the amount and composition of the exhaust gases from motor vehicles in relation to the ventilation of vehicular tunnels. Report of the New York State Bridge and Tunnel Commission to the Governor and Legislature of the State of New York; Legislative Document No. 64, 1921, pp. 97-140.
Fieldner, A. C., Straub, A. A., and Jones, G. W., Gasoline losses due to incomplete combustion in motor vehicles: Bureau of Mines, Report of Investigations No. 2225, March, 1921, 19 pp.

Paper presented at the Semi-Annual Meeting of American Society of Heating and Ventilating Engineers, Chicago, Ill., May, 1923. Submitted by permission of the Director, Bureau of Mines.

TABLE 1. AIR REQUIRED TO DILUTE AUTOMOBILE EXHAUST GASES TO SAFE PROPORTIONS IN AIR BREATHED

(Car tested under summer conditions unless otherwise stated)

Number of Cars Tested	Type of Car[1]	Speed miles per hour	Average carbon monoxide in exhaust gas per cent	Cu. ft. CO per hr., per car (at 65° fahr. 29.92" Hg)	Cu. ft. per min. of ventilating air per car required to dilute CO to 1 part per 10 000 (safe for indefinite period)
		Engine Idling			
5	5P	Car standing	7.3	39.2	6533
7[2]	7P	"	6.3	35.3	5883
5[2]	1½T	"	7.1	31.0	5167
4	1½-3	"	2.4[3]	13.2[3]	2200[3]
3	3½-4½	"	8.8	66.4	11070
6	5T	"	6.8	50.5	8417
		Engine Racing			
5	5P	Car standing	7.1	76.9	12820
7[2]	7P	"	7.8	137.0	22830
5[2]	1½T	"	7.7	67.9	11320
4	1½-3	"	4.7	55.8	9300
3	3½-4½	"	8.6	158.2	26370
6	5T	"	7.5	105.2	17530
		Average Load and Level Grade			
11[2]	5P	3	6.7	36.3	6050
7[2]	7P	3	6.6	53.1	8850
5[2]	1½T	3	7.8	47.4	7900
22	1½-3	6	7.3	67.6	11270
16	3½-4½	6	7.8	92.6	15430
18	5T	6	7.5	110.3	18380
9	5P	10	7.8	44.1	7350
16	7P	10	8.3	75.5	12580
7	1½T	10	7.6	58.7	9783
20	1½-3	10	7.5	104.0	17330
13	3½-4½	10	7.5	147.6	24600
12	5T	10	6.9	151.6	25270
9	5P	15	6.8	57.9	9650
6	7P	15	8.6	111.8	18630
7	1½T	15	6.7	77.2	12870
15	1½-3	15	6.2	103.9	17320
5	3½-4½	15	4.8	131.2	21870
		Full Load and 3% Grade Upward			
7[2]	7P	3	8.4	88.5	14750
5[2]	1½T	3	9.5	65.1	10850
22	1½-3	6	6.9	114.3	19050
16	3½-4½	6	6.9	148.1	24680
18	5T	6	6.1	163.3	27220
6	7P	10	8.9	117.2	19530
7	1½T	10	7.3	86.3	14380
22	1½-3	10	6.7	149.6	24930
16	3½-4½	10	6.1	88.5	14750
18	5T	10	4.9	199.3	33220
13	7P	15	8.3	146.7	24450
12	1½T	15	6.9	117.7	19620
15	1½-3	15	5.4	130.1	21680
16	3½-4½	15	4.1	181.7	30280
		Accelerating up 3% Grade with Full Load			
7[2]	7P	0 to 15	6.5	163.6	27270
5[2]	1½T	0 to 15	5.4	65.9	10980

[1] Types are designated thus: 5P and 7P represent 5 and 7 passenger cars, 1½T, 1½-3, 3½-4½ and 5T represent trucks of the respective tonnages.
[2] Tested under winter conditions.
[3] These averages are undoubtedly too low to represent correctly the type of trucks. All cars were tested with carburetors set as submitted by the car owners. These four happened to be set for lean mixtures.

the amounts of pure air needed to dilute the exhaust gases below the toxic concentrations and remove them.

PHYSIOLOGICAL EFFECTS OF CARBON MONOXIDE

The effects of inhaling carbon monoxide, described in detail in the literature to which reference is given, are as follows:

Exposure to 0.4 per cent (40 parts per 10,000) of carbon monoxide in air for 1 hour may kill a man.[4]

A man doing sedentary work as when driving an automobile may breathe 0.04 per cent (4 p.p. 10,000) up to one hour without ill effects.[5]

Exposure to 0.025 per cent (2½ p.p. 10,000) causes mild to moderate effects in one hour of strenuous exercise.[6]

Continuous breathing of 0.01 per cent (1 p.p. 10,000) produces no noticeable ill effects on adults in good health.[7]

DISPERSION OF EXHAUST GAS

Undiluted exhaust gas as discharged from the motors is highly poisonous, and if breathed for a few minutes without admixture of air would quickly cause unconsciousness and death. However, the gas mixes with some air immediately upon discharge due to its high velocity and kinetic energy, and correspondingly low pressure. Automobiles in motion disturb the air about them according to their speed and the currents and eddies cause further dilution. Exhaust gas cooled to atmospheric temperature is somewhat heavier than air, but the high temperature at which it is emitted lightens it considerably so that it rises and in so doing becomes again further diluted. The net result of all these effects is an immediate dilution of discharged gas with air and then a tendency to rise to the ceiling. Thereafter the gases may escape upward through elevator shafts through cracks or pores in the ceiling, or outward through walls, windows or doors or may mix further with the room air due to the normal air currents therein, or by diffusion, of which the latter is a molecular process and relatively slow.

The tendency of the warm gases to rise has sometimes caused headache and sickness among persons in upper stories of buildings, while persons in the lower rooms where the gas is liberated felt no effects. Similar instances have occurred when stationary internal combustion engines in basements have developed leaks in the exhaust pipes.

When doors and windows are open in summer, there is usually enough natural ventilation to remove poisonous gases from rooms or buildings,

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[4] Henderson, Yandell, Carbon monoxide poisoning: *Journal, Amer. Med. Assn.*, Vol. 67, Aug. 19, 1916, pp. 580-3.

[5] Henderson, Y., Haggard, H. W., Teague, M. C., Prince, A. L., and Wunderlich, R. M., Physiological Effects of Automobile Exhaust Gas and Standards of Ventilation for Brief Exposures: *Journal Ind. Hyg.*, Vol. 3, August, 1921, pp. 79-92 and 137-146.

[6] Sayers, R. R., Meriwether, T. V., and Yant, W. P., Physiological Effects of Exposure to Low Concentrations of Carbon Monoxide: Bureau of Mines Report of Investigations No. 2338, March, 1922, 6 pp.

[7] Sayers, R. R., Yant, W. P., and Fulton, W. B., Unpublished experiments of U. S. Bureau of Mines.

so that occupants are not affected. In winter, however, when buildings are closed, special provision for removing contaminated air is desirable and oftentimes necessary. It is easily shown that carbon monoxide must escape either by natural ventilation processes or by specially installed methods for the purpose, for the discharge from a single automobile can in one or two hours time create a deadly atmosphere in a large room which is completely closed and unventilated. The natural processes change with outdoor wind or weather conditions so that a loosely constructed building which usually has satisfactory ventilation may develop very unhygienic atmospheres on calm days.

AIR REQUIRED TO DILUTE AUTOMOBILE EXHAUST GAS TO SAFE LIMITS

Table 1 states the average percentage and total volume per hour of carbon monoxide gas discharged by each of various types of passenger automobiles and trucks under full rated load and average of full and no load. The last column gives the volumes of pure air needed to dilute the carbon monoxide per car to 1 part in 10,000, which does not harm adults if breathed indefinitely; this dilution is recommended for garages and automobile shops.

So far as available the data are from automobiles operated in summer since it was found that slightly greater volumes of carbon monoxide are produced then due to higher volatility of gasoline and richer mixtures at summer temperatures.

The average figures in the table may be used to calculate ventilation requirements of commercial garages where various types of cars come and go. Individual cars of any class with carburetors set for extremely lean or rich mixtures may, however, produce respectively only a fraction of the average volumes of carbon monoxide or up to about 100 per cent more, requiring each for itself alone, correspondingly less or more ventilating air. Thus the data for 1½ to 3 ton trucks with idling engines, showing 2.4 per cent carbon monoxide in the exhaust is undoubtedly somewhat low for the class in general. All cars were tested with carburetors set as submitted by the owners and these four 1½ to 3 ton trucks all happened to be set lean. The average carbon monoxide percentage for each class of cars, under all conditions of test, was: 5 passenger cars, 6.4; 7 passenger cars, 7.1; trucks up to 1½ tons capacity, 6.9; trucks, 1½ to 3 tons inclusive, 5.8; trucks 3½ to 4½ tons, 6.8; trucks, 5 tons and over, 6.5.

SUMMARY

Data have been presented on the volumes of carbon monoxide gas discharged from various automobile engines, and on the volumes of air necessary to dilute the carbon monoxide to 1 part in 10,000 of air which is recommended as a safe concentration for workers in garages and shops. Natural ventilation is usually sufficient in summer when doors and windows are open, but special provision for removing the poisonous gas in winter when buildings are closed is sometimes desirable.

It may not be amiss to repeat here that automobile engines should never be operated in small garages when the doors and windows are closed; many deaths have resulted from such practice.

DISCUSSION

P. J. DOUGHERTY: I would like to inquire about the physiological reactions of carbon monoxide on the erythrocytes (red corpuscles) the leukocytes (the white corpuscles) or the hemoglobin?

S. H. KATZ: Carbon monoxide attacks only the hemoglobin, which is the coloring matter in the red corpuscles of the blood and which normally transports oxygen from the air in the lungs to the tissues throughout the body. If carbon monoxide is present in the air breathed it also combines with the hemoglobin in a manner similar to the oxygen but with an "affinity" or "avidity" about 250 to 300 times that of the oxygen. As a result of this the carbon monoxide when breathed continuously and even in low concentrations, tends to accumulate in the blood and thus to reduce the oxygen carrying capacity. The person becomes suffocated for want of oxygen, although he may be surrounded by it in abundance. In concentrations of 2 per cent of carbon monoxide death may come almost as quickly as from drowning. At lower concentrations the accumulation is correspondingly slower.

When the person is removed from the carbon monoxide containing atmosphere to pure air, which is the first essential in giving aid or treatment, the oxygen slowly displaces the carbon monoxide from the blood but the expulsion is very much slower than the accumulation. If breathing is shallow or has stopped, the Schaeffer prone pressure method of artificial respiration should be administered to bring resuscitation.

It has been found that headaches, nausea or weakness usually develop when the blood is 30 per cent saturated. The activity of the person greatly accentuates this, if he is working vigorously the effects are rapidly brought about. At 50 to 60 per cent a person loses consciousness and at 75 to 80 per cent death usually results. If the accumulation of carbon monoxide is rapid there are no subjective symptoms. Consciousness is lost without any warning. In low concentrations of carbon monoxide the blood may become only partly saturated and then remains without increase so that no noticeable ill effects are produced. The concentration of one part of carbon monoxide in 10,000 parts of air is recommended as a safe upper limit for garages.

I do not know offhand what percentage of the blood will be saturated with carbon monoxide when in equilibrium with this concentration, but Mr. Yant can probably tell us.

W. P. YANT: When hemoglobin is exposed to both oxygen and carbon monoxide there is a competition for the hemoglobin by both gases, the competitive force of each being the concentration or partial pressure [P] of each gas times its affinity [A] or $(P \times A)$ and the amount of hemoglobin that will combine with each will be in the ratio of the competing forces.

$$\frac{Pco \times Aco}{Po_2 \times Ao_2} = \frac{CO\ Hb}{O_2\ Hb}$$ Then according to the rules of proportion

$$\frac{Pco \times Aco}{(Po_2 \times Ao_2) + (Pco \times Aco)} = \frac{CO\ Hb}{O_2Hb + CO\ Hb}$$ and

$$\text{Per cent of } CO\ Hb = \frac{Pco \times Aco \times 100}{(Po_2 \times Ao_2) + (Pco \times Aco)}$$

Using this formula and the values for the different terms as given:
$Aco = 300$.
$Ao_2 = 1$, on basis that $Aco = 300$.
$Po_2 = 1{,}500$ pp 10,000, the approx. amount of O_2 in pulmonary air.
$Pco =$ the concentration CO in question, 1 pp 10,000 in this case.

$$\text{Per cent of } CO\ Hb = \frac{1 \times 300}{(1{,}500 \times 1) + (1 \times 300)} \times 100 = 16.6$$

Restating this, a man at rest, breathing air containing 1 pp 10,000 CO will have 16.6 per cent of his hemoglobin combined with CO when equilibrium occurs. This figure will vary a little with different individuals but for all practical purposes it can be considered as a good average. Further it is not likely that the exposure would continue until the theoretical equilibrium is attained since that would take a long time.

No. 671

FURTHER OBSERVATIONS IN DUST DETERMINATIONS

By Margaret Ingels[1], Pittsburgh, Pa.

Member

An important investigation has been made in the further development of the Anderson and Armspach dust determinator. This investigation is a microscopic study of the dusts collected on filter papers when the air containing the dust is passed through papers in series. Three phases of this problem are presented in this report.

1. The amount of dust not collected by the first paper but collected by the second, third, fourth or fifth in series.
2. The nature in which the paper collects the dust.
3. The size of the particles collected.

Two kinds of dust were passed through the filter papers. The first dust which in this report is called "room dust," was that found in Pittsburgh air, floating naturally and is typical of this industrial city. This dust was used as representative of that in a naturally dusty atmosphere and the size of particles would be comparable to that found in any air containing dust. The second dust which in this report is called "tobacco smoke dust" is dust from cigarette smoke introduced into the air. Tobacco smoke dust was used because most methods for indicating this dust run very low in efficiency. Tobacco smoke dust acts as a combination of gas, liquid and dust so that the efficiency for any dust remover may be expected to be lower than that for air containing dust only. As the breathing apparatus of the human system is such an excellent *air washer* the test with tobacco smoke dust was divided into two parts, inhaled, and not inhaled smoke. The inhaled smoke was drawn into the lungs, then introduced into the air. The not inhaled smoke was introduced directly into the air at the intake of the apparatus.

In each of the tests five Whatman chemical filter papers No. 30 were placed in the filter frame of the Anderson and Armspach dust determinator and air drawn through them in series at the rate of ½ c.f.m. As the resistance gage for the filter medium has a maximum reading of 14

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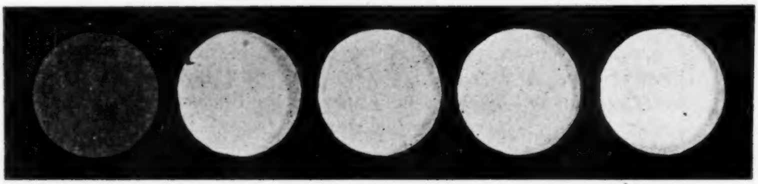
in., and the total resistance of these filters is approximately 30 in., the gage was disconnected during the tests. The amounts of dust collected were not measured in any way.

Specimens for microscopic study of each filter paper in each test were made as follows: a portion of the exposed area of each paper was placed on a microscopic slide, the slide being less than 0.90 m.m. thick to allow for light transmission; the sample thus obtained was immersed in glycerine and covered with a cover glass.

Two degrees of magnification were used in the microscopic study of the collected dusts. The usual dust estimation made by the *count* method, is done under a magnification of 200 diameters, so that this magnification is used in these studies. It is believed that at this magnification the



1st Ply 2nd Ply 3rd Ply 4th Ply 5th Ply

FIG. 1. ROOM AIR DRAWN THROUGH FILTER PAPERS IN SERIES

particles are not divided into their primary parts, and that some particles are not seen at all. The highest magnification possible is desirable to see the smallest particles, as the smallest particles are the hardest to remove by artificial means, float the longest, and still may carry a dreaded germ, so that the supermagnification of 1000 diameters is used.

It must be remembered when looking at the micro-photographs which are presented illustrating this report, that all the dust particles appearing on the field are not in focus. The greater the magnification the greater the impossibility to get a large number of particles in the same focus. Those out of focus will appear as a black blur.

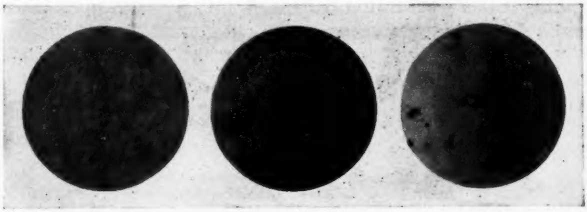
In the first test, air containing room dust was passed through five filter papers; the actual filter papers are shown in Fig. 1. The *color scale* here is interesting but does not give all that can be known. The first phase of the problem is roughly answered in that the first paper catches all the dust.

Photographs of the first and second filter papers, magnified 200 diameters are shown in Figs. 2 and 3. They show that the dust collected on the first paper is so great in comparison to that collected on the second, that 100 per cent efficiency may still be assumed for the first paper. At this magnification the second phase of the problem seems possible to answer. It can be seen how the large particles are caught first; the smaller imbedding themselves between the fibres of the paper. The third phase of the problem cannot be answered for, at this magnification, the size of secondary particles cannot be estimated. That which here may look to be a large particle may be several hundred particles stuck to-

gether by impact. The actual field in Figs. 2 and 3 is 0.381 m.m. in diameter.

When studying a field at a magnification of 1000 diameters it was found that the dust was collected through a depth of 0.18 m.m. It is

x 200 x 200 x 1000

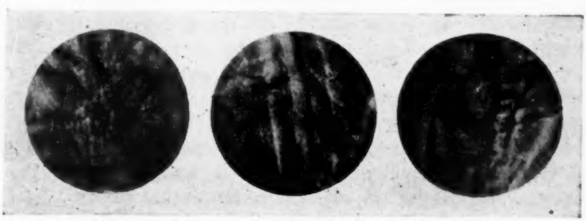


1st Ply of Paper Fig. 2 2nd Ply of Paper Fig. 3 1st Ply—Top Plane Fig. 4

x 1000 x 1000 x 1000

1st Ply—Fibre Plane Fig. 5 2nd Ply—Top Plane Fig. 6 2nd Ply—Fibre Plane Fig. 7

x 1000 x 1000 x 1000



3rd Ply—Fibre Plane Fig. 8 4th Ply—Fibre Plane Fig. 9 5th Ply—Fibre Plane Fig. 10

ROOM DUST DEPOSITED ON VARIOUS PLY FILTER PAPERS AS VIEWED UNDER MICROSCOPE

impossible to focus on all particles through this depth so a photograph was taken of the top plane and of the lower or fibre plane. An intense transmitted light was used, so the dust particles out of focus were trans-

parent. Figs. 4 and 5 are photographs of the same field, No. 4 being focused on the top plane, No. 5 focused on the fibre plane. The total dust indicated by these two photographs is less than the total collected, for a large number of particles fall between these two planes and are not in the field of vision of either. The actual field is 0.076 m.m. in diameter.

The photographs of the two planes for the second paper are shown in Figs. 6 and 7. The view of the top plane indicates that no large particles

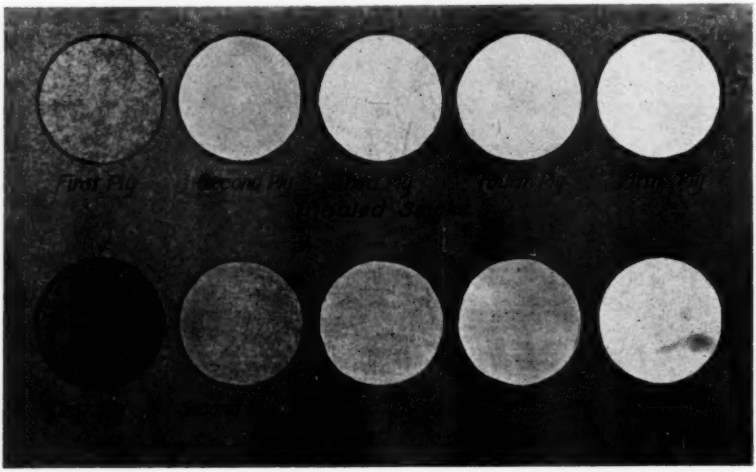


FIG. 11. CIGARETTE SMOKE

of dust passed through the first paper. The view of the fibre plane shows that very few small particles are collected on the second paper. It is not possible to determine what per cent of the whole is collected on this paper as the weight is too small for measurement and the count method is inaccurate. The dust collected by the second paper in comparison to that collected by the first is less than the per cent error allowable for the most careful and accurate scientific research.

Figs. 8, 9 and 10 show photographs of the third, fourth and fifth papers in series respectively. Dust that is not caught by the first or second paper is not caught on any of the remaining papers.

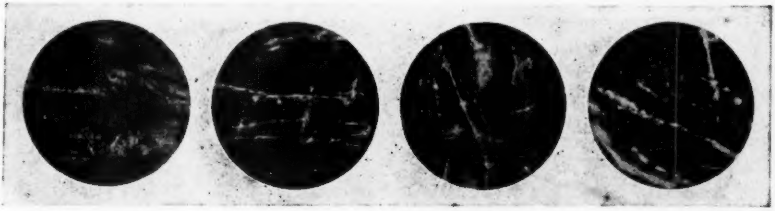
A comparison may be made between Fig. 2 and Figs. 4 and 5. The ratio of the areas of Fig. 2 and Figs. 4 and 5 is 5 to 1, therefore the ratio of the dust indicated should be the same. That is, Fig. 2 should indicate five times the dust indicated by the sum of Figs. 4 and 5. This it does not do because the magnification in Fig. 2 is not high enough to show the smallest particles.

The supermagnification shows how little dust gets through the first filter paper. The manner in which the dust is collected can be seen, and

the sizes of the primary particles may be observed and estimated in microns.

Tobacco smoke dust filters are shown in Fig. 11. It is interesting to note that all the dust of inhaled tobacco smoke is apparently caught on the first paper, while some dust of the not inhaled tobacco smoke is shown on all five papers. One explanation of this may be that the smallest particles which get through the first paper when the smoke is not inhaled are removed by the pulmonary system when the smoke is inhaled. A second explanation may be that the dust of inhaled smoke is saturated with moisture and the smaller particles are thus made larger and caught by the first paper. The first explanation is more probable for all smoke

x 1000 x 1000 x 1000 x 1000



Inhaled	Inhaled Separate Fibres	Not Inhaled	Not Inhaled Separate Fibres
Fig. 12	Fig. 13	Fig. 14	Fig. 15

TOBACCO SMOKE DUST INHALED AND NOT INHALED

from cigarettes is saturated with moisture formed by oxidation. Fig. 12 is a photograph of the first filter paper when inhaled cigarette smoke was introduced into the air. Fig. 13 shows two or three fibres torn from the main sample and photographed. Each of these photographs are with a magnification of 1000 diameters.

Figs. 14 and 15 are photographs of the paper and its fibres when the cigarette smoke was not inhaled. There is not the same amount of dust collected in Figs 12 and 13 as in Figs. 14 and 15, for there was not the same amount of smoke liberated. The cigarette smoke dust appears as colloidal particles and is collected along the surface of the fibre.

Another test was made to learn how dust suspended in a liquid and magnified according to the practice used in so many dust estimations would appear under supermagnification.

A heavy fine dust suspended in water was desired for a specimen. Cigarette smoke not inhaled was bubbled through distilled water. A drop of this water was put on a microscopic slide and the water allowed to evaporate. The dust deposit left on the slide was submerged in glycerine and a cover glass put over it. Four pictures were made of the same field under the microscope and these photographs are shown in Figs. 16, 17, 18 and 19. In Figs. 16 and 18 the fields are magnified diameters

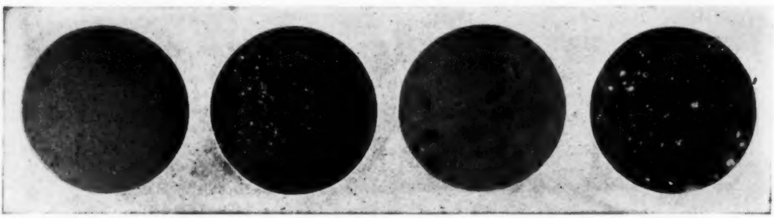
and 1000 diameters respectively. Each time ordinary transmitted light was used. In both photographs the light is not sufficient to give a clear outline to the particles. Fig. 16 shows a typical field from which dust counts are made and it can easily be seen that the possible error in counting the particles may be large.

Figs. 17 and 19 are pictures of the same field with intensified light as Figs. 16 and 18 respectively.

By using intense light the particles stand out with a clear outline even at the 200 diameters magnification. The field shown in Fig. 16 can be more easily fancied as dust particles collected on a white field but, for counting, the intensified light reduces the chance of error.

In the fields where the light is different and the magnification is 1000 diameters the same difference in a distinct outline of dust particles is seen as in those with 200 diameters magnification.

x 200 x 1000 x 200 x 1000



Ordinary Light Fig. 16 | Intense Light Fig. 17 | Ordinary Light Fig. 18 | Intense Light Fig. 19

TOBACCO SMOKE DUST SUSPENDED IN WATER

At the high degree of magnification some of the particles are still seen to be secondary, and in all probability no matter what degree of magnification were possible some particles would still be secondary.

J. S. Owens, of London, England, gives as the most satisfactory test for his Jet dust counting apparatus an experiment in which he assumes a chemical filter paper as 100 per cent efficient.

The tests with the tobacco smoke dust show the extreme sensitiveness of the Anderson and Armspach dust determinator and the room dust tests show its high efficiency.

When the amount of dust is too small to weigh, the particles too small to count, the Anderson and Armspach dust determinator indicates even the smallest degree of dustiness by an increase in resistance of the filter.

Acknowledgment is made to Dr. Reinhardt Thiessen of the U. S. Bureau of Mines for his co-operation and the use of his equipment for the microscopic work and photographs. (*Note:* In reproduction of photographs the reduction is two-thirds.)

No. 672

FURTHER STUDY OF PHYSIOLOGICAL REACTIONS

By W. J. McConnell[1] (Non-Member)

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It is frequently said that ventilation is a subject concerning which engineers and physicians are by no means all agreed. A review of the literature justifies the assertion and further analysis indicates that the basis of such diversity apparently rests upon failure to discriminate between facts, opinions, bias and prejudice.

Present ventilation standards are largely empirical and, although based upon the best opinions of leading authorities in this field of work, still, they are lacking in scientific exactness. It is therefore quite natural for different groups of individuals to be governed by their own particular experiences and to formulate their own standards of ventilation.

With the discovery of more fundamental facts and a study of the combined viewpoint of all essential factors concerned in the problem of ventilation, these differences will cease to exist, because facts once established permit investigation of them from every angle.

The results of the investigations into the effects of atmospheric-conditions now being conducted at the experimental laboratory of the U. S. Bureau of Mines jointly by a body of physicians and engineers in search of scientific knowledge of the demands of ventilation for the promotion of health, comfort and efficiency of the human body, will undoubtedly determine a logical standard of ventilation. The once accomplished, marks a definite advance in the science of ventilation.

Again, the question arises as to the best method available for measuring the endurance, consistent with health, of those exposed to abnormal environmental conditions, and also whether it is possible to differentiate between a disinclination to do work under unusual

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temperature conditions and actual inability. The results of these studies should prove very useful in attaining this information, because any conclusions drawn from these researches are based, not upon animal experimentation, but upon physiological reactions of human subjects under accurately controlled conditions.

These studies will likewise supply the medical profession with information relative to the altered processes within the body, such as

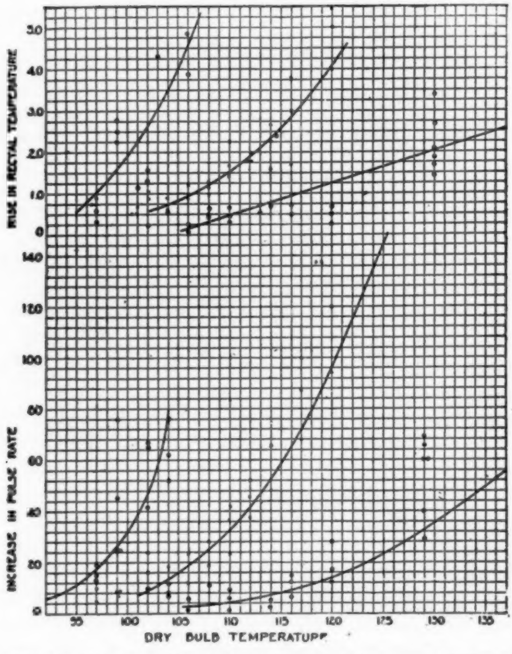

FIG. 1. INCREASE IN PHYSIOLOGICAL REACTION MEASURES ACCORDING TO DRY BULB TEMPERATURE, AFTER ONE HOUR OF EXPOSURE

changes in the metabolic rate, in the hydrogen, in the concentration of the blood and in other vital processes of the body which may occur incident to exposure to various temperatures, humidities and velocities of air.

In contemplating future work in certain industrial processes where a certain atmospheric condition must be maintained, these investigations will indicate to industrial managers what they can expect in the way of efficiency and tolerance of those so exposed. They will likewise indicate the temperature conditions at which fans are desirable; also when fans will no longer mitigate the heat and finally when an increase in air currents may even exaggerate the discomfort.

Engineers will have available information useful as a guide in measuring atmospheric conditions and, knowing the demands of ventilation, they can devise means to provide human beings, living within buildings, with air adapted to their use.

The results of these experiments will serve the purpose of informing air conditioning engineers to what extent optimum equivalent conditions can be maintained by substitution of humidity for temperature, and how far air circulation affects or ameliorates excessive temperature-humidity combinations.

These studies furthermore will furnish very practical applications in the mining industries of the country. Excessive temperature and humidity predominate in most mines and conditions found in many of these have been duplicated in our experiments and the effects on human subjects have been carefully recorded. These records demonstrate the physiological changes which take place in the body, and indicate the length of time human beings can tolerate the existing conditions of temperature and humidity. With this information at hand steps can be taken to combat any exigencies of environment.

These few applications suggest others, and on completion of each phase of the investigation still others will present themselves.

A report of the first phase of the investigation namely, Some Physiological Reactions to High Temperatures and Humidities in Still Air, was published in the March 1923 JOURNAL, This report included a large amount of data and certain definite conclusions.

The second phase of the investigation, namely, Physiological Reactions to High Temperatures and Humidities with Moving Air, is being actively pursued by the three services. A large amount of data has already been collected which point to some very interesting results. Sufficient facts are not yet available however to warrant definite conclusions. A report on this phase of the work may be expected at the next meeting of the Society.

Discussion, illustrations and conclusions in the earlier reports were largely drawn from single sets of observations rather than from a statistical study of all data.

It is of interest to put together such data as are homogeneous to get, if possible, a composite picture of these physiological changes under various dry bulb, wet bulb and effective temperature and finally under different relative humidities.

It will first be attempted to show changes in pulse rate, body temperature, and systolic pressure, under the different temperatures and humidities, graphically. The data are treated by the method of least squares, and the curves plotted from the resulting equations. This best fitting curve is plotted on the same sheet with the independent observations so that the general trend of the observations and their deviations may be seen at each point in the curve. The curves are plotted for changes taking place during 1 hr. of exposure. When the time was less than 1 hr., but not less than 40 min., a straight

line interpolation is made to obtain an hour value. There are comparatively few of these interpolations, and some of them give the excessive values noted in the graphs.

Rise in rectal temperature after 1 hr. of exposure is plotted against D. B. temperature in the upper set of curves in Fig. 1. The three

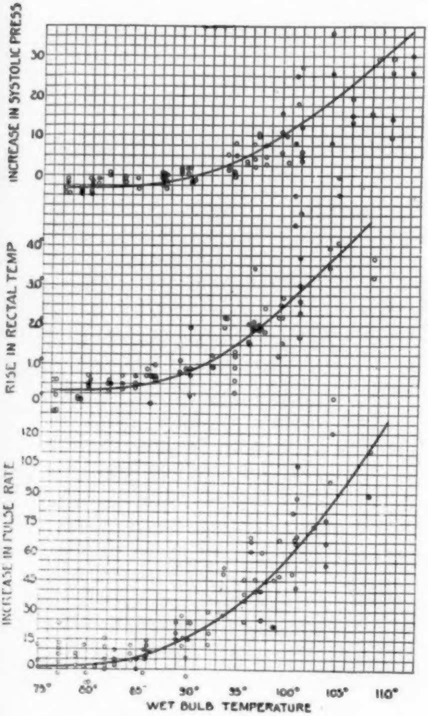

FIG. 2. INCREASE IN PHYSIOLOGICAL REACTION MEASURES ACCORDING TO WET BULB TEMPERATURE, FOR DIFFERENT HUMIDITIES, AFTER ONE HOUR EXPOSURE

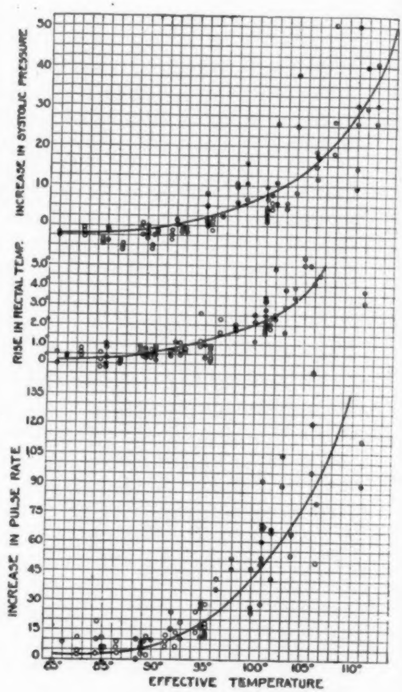

FIG. 3. INCREASE IN PHYSIOLOGICAL REACTION MEASURES ACCORDING TO EFFECTIVE TEMPERATURE AFTER ONE HOUR OF EXPOSURE

curves from left to right are for 100 per cent, 60 per cent and 30 per cent relative humidity respectively. Increase in pulse rate per hour is similarly plotted against D. B. temperature in the lower set of curves Fig. 1, for the three different relative humidities.

In Fig. 2 the changes in physiological measures are plotted against wet bulb temperature without regard to dry bulb temperature or relative humidity. It will be seen that the observations show considerable variation and that the rate of increase of physiological

reactions is much greater at high wet bulb temperatures than at low. Between 75 and 80 deg. W. B. there is practically no change in the body reactions. As the wet bulb temperature increases the reactions increase slowly, until at 90 deg. a more rapid increase begins. At 104 deg. W. B. the pulse rate is nearly double the normal rate.

Rise in rectal temperature and change in systolic pressure show similar characteristics.

The same physiological reactions are plotted against effective temperature in Fig. 3. Although the general characteristics of these curves are similar to those in Fig. 2, a close study of the two sets of curves will disclose the fact that the physiological reactions follow the scale of effective temperature better than the wet bulb scale. Especially in the plots for rises in rectal temperature this fact is very apparent.

To demonstrate clearly the superiority of effective temperature over the wet bulb temperature as a measure of body reactions, further deductions may be made from the curves. In Fig 1, the intersection of a horizontal line with the three curves for various relative humidity gives three distinct atmospheric conditions which produce the same body reactions. If a horizontal line is drawn through an increase in pulse rate of 30 beats, it intersects the 100 per cent relative humidity curve at 99.8 deg. D. B. temperature, the 60 per cent relative humidity curve at 109.5 deg D. B. and the 30 per cent curve at 128.0 deg. D. B. The wet bulb temperatures of these three equivalent conditions are 99.8, 95.5 and 95.0 deg. respectively and the effective temperatures are 99.8, 97.8 and 99.9 deg. respectively. It will be seen that the variation of the three wet bulb temperatures is much greater than that of the effective temperatures.

If a horizontal line representing a rise in rectal temperature of two degrees is taken, the three conditions at which this line intersects the three curves at 100.2 deg. D. B., 100.2 deg. W. B. and 100.2 deg. E. T. for the 100 per cent curve, 112.4 deg. D. B., 98.0 deg. W. B. and 100.1 deg. E. T. for the 60 per cent curve and 129.5 deg. D. B., 96. 2 deg. W. B. and 100.8 deg. E. T. for the 30 per cent relative humidity curve. Again the effective temperatures for the conditions giving the same physiological reactions are practically the same while there is considerable difference in the wet bulb and a very great difference in the dry bulb temperatures for the same body reactions.

Many interesting facts are brought to light by a study of these changes in pulse rate, systolic pressure, and body temperature, as affected by different conditions of atmospheric temperature and humidity. A study is first made of the correlation between physiological reactions and wet and dry bulb temperature, respectively. The correlation between these reactions and effective temperature will then be taken up. It should be kept in mind that perfect direct correlation is denoted by 1.000, perfect inverse correlation by — 1.000, and lack of correlation by 0.000.

In Table 1, there are given the zero order correlations for pulse rate after one hour of exposure.

TABLE I. CORRELATIONS FOR PULSE RATE

	Wet Bulb	Dry Bulb	Humidity	Pulse Rate
Pulse Rate	0.8200 ± 0.031	0.576 ± 0.086	0.7160 ± 0.042	
Wet Bulb		0.0548 ± 0.087	0.8880 ± 0.020	
Dry Bulb			—0.3490 ± 0.076	
Stand. Deviation	8.9	5.2	26.1	19.2

The correlation between pulse rate and wet bulb reading is relatively high, 0.820, that between pulse rate and relative humidity, lower, 0.716, while that between pulse rate and dry bulb is very low, 0.058. It should be noted, however, that these correlations are not net, because in each case the other factors have not been eliminated. For example, the correlation between pulse rate and wet bulb, 0.820, may be due partially to the combined effects of the other factors. Partial correlations are resorted to in order to eliminate the effect of the other factors, that is, to hold them constant at their mean values. When this is done for each factor in turn the first order correlations given in Table 2 are the result.

TABLE 2. PARTIAL CORRELATION FOR PULSE RATE

Correlation between	Held Constant	Correlation $r =$
Pulse rate and wet bulb	Dry bulb	0.8195 ± 0.031
Pulse rate and wet bulb	Humidity	0.5738 ± 0.086
Pulse rate and dry bulb	Wet bulb	0.0222 ± 0.087
Pulse rate and dry bulb	Humidity	0.4703 ± 0.069
Pulse rate and humidity	Wet bulb	—0.0275 ± 0.087
Pulse rate and humidity	Dry bulb	0.7867 ± 0.032

The correlation between pulse rate and wet bulb is still relatively high when each other factor in turn is held constant, being 0.82 and 0.574, respectively. That for pulse rate and dry bulb is reasonably high when humidity is held constant, and that for pulse rate and humidity is relatively high when dry bulb is held constant. The second order coefficients, which are net as far as the other factors are concerned, are given in Table 3.

TABLE 3. SECOND ORDER COEFFICIENT FOR PULSE RATE

Correlation between	Held Constant	r value
Pulse rate and wet bulb	Dry bulb and humidity	0.3660 ± 0.076
Pulse rate and dry bulb	Wet bulb and humidity	—0.0020 ± 0.087
Pulse rate and humidity	Wet bulb and dry bulb	—0.0164 ± 0.087

The only net correlation of any size, is that between pulse rate and wet bulb, 0.366. The correlation between pulse rate and each of the other factors is small, as well as inverse, and apparently negligible. It would thus appear that the wet bulb reading is more important than the dry bulb or humidity. The total correlation, that is, the combined effect of these three factors upon pulse rate is, $R\ 1.234 = 0.839$.

A similar study is made for rectal temperature. The zero order coefficients are shown in Table 4.

TABLE 4. CORRELATIONS FOR RECTAL TEMPERATURES

	Wet Bulb	Dry Bulb	Humidity	Rectal Temperature
Rectal Temp.	0.7310 ± 0.041	0.0076 ± 0.087	0.6600 ± 0.050	
Wet Bulb		0.0548 ± 0.087	0.8880 ± 0.020	
Dry Bulb			—0.3491 ± 0.076	
Stand. Dev.	8.9	5.2	26.1	1.6

Again the correlation is significant for wet bulb and humidity, and low for dry bulb. The first order coefficients are as given in Table 5.

TABLE 5. FIRST ORDER COEFFICIENT FOR RECTAL TEMPERATURES

Correlation between	Held Constant	r value
Body Temperature and wet bulb	Dry bulb	0.7299 ± 0.041
Body Temperature and wet bulb	Humidity	0.4197 ± 0.073
Body Temperature and dry bulb	Wet bulb	—0.0476 ± 0.087
Body Temperature and dry bulb	Humidity	0.4286 ± 0.071
Body Temperature and Humidity	Wet bulb	0.0347 ± 0.087
Body Temperature and Humidity	Dry bulb	0.7178 ± 0.043

Here again relatively high correlations are found between body temperature and wet bulb where dry bulb is constant, and between body temperature and humidity where dry bulb is held constant. The correlation, as before, is significant between body temperature and wet bulb with humidity constant, and between body temperature and dry bulb, with humidity constant. The net correlations are as given in Table 6.

TABLE 6. SECONDARY COEFFICIENTS FOR RECTAL TEMPERATURES

Correlation between	Held Constant	r value
Rectal temperature and wet bulb	Dry bulb and humidity	0.2109 ± 0.084
Rectal temperature and dry bulb	Wet bulb and humidity	—0.0340 ± 0.087
Rectal temperature and humidity	Wet bulb and dry bulb	—0.0101 ± 0.087

Here again the only correlation worthy of note is that between rectal temperature and wet bulb. It is not as high as that for pulse rate. The total correlation for body temperature is, $R\ 1.234 = 0.762 \pm 0.038$.

The correlations for systolic pressures are given in Table 7.

TABLE 7. CORRELATION FOR SYSTOLIC PRESSURE

	Wet bulb	Dry bulb	Humidity	Systolic
Systolic	0.8012 ± 0.029	0.2870 ± 0.074	0.7075 ± 0.035	
Wet bulb		0.2785 ± 0.075	0.9222 ± 0.013	
Dry bulb			—0.1083 ± 0.080	
Stand. Dev.	11.8	5.1	30.1	15.2

Again wet bulb and humidity outweigh dry bulb in these zero order correlations. The first order correlations appear in Table 8.

TABLE 8. FIRST ORDER COEFFICIENTS FOR SYSTOLIC PRESSURE

Correlation between	Held Constant	r values
Systolic and wet bulb	Dry bulb	0.7840 ± 0.032
Systolic and wet bulb	Humidity	0.5444 ± 0.056
Systolic and dry bulb	Wet bulb	0.1111 ± 0.079
Systolic and dry bulb	Humidity	0.6312 ± 0.050
Systolic and humidity	Wet bulb	—0.1356 ± 0.079
Systolic and humidity	Dry bulb	0.7755 ± 0.032

What has been said about pulse rate and rectal temperature, may be repeated for systolic pressure as far as first order coefficients are concerned.

The net correlations for systolic pressure are given in Table 9.

TABLE 9. SECOND ORDER COEFFICIENT FOR SYSTOLIC PRESSURE

Correlation between	Held Constant	*r* values
Systolic and wet bulb	Dry bulb and humidity	0.2299 ± 0.075
Systolic and dry bulb	Wet bulb and humidity	-0.1200 ± 0.079
Systolic and humidity	Wet bulb and dry bulb	-0.1429 ± 0.078

Again there is a better correlation between systolic and wet bulb readings than between systolic and either of the other factors here considered. In fact, these other correlations are inverse, when they stand alone in relation to systolic pressure. The total correlation is: $R\ 1.234 = 0.799 \pm 0.029$.

In every correlation so far studied, the coefficient is at least significant when the wet bulb reading is a factor.

A study will next be made of the correlation between physiological reactions and wet bulb and effective temperatures, respectively. In this connection correlation ratios will be used instead of correlation coefficients. The results are shown in Table 10.

TABLE 10. CORRELATION RATIOS BETWEEN PHYSIOLOGICAL CHANGES AND BOTH WET BULB AND EFFECTIVE TEMPERATURES

Temperature	Rectal Temperature	Pulse Rate	Systolic Pressure
Wet Bulb	0.930	0.906	0.752
Effective Index	0.975	0.961	0.880

In every case the correlation ratio is higher when the physiological changes are based upon effective temperatures. This means that these indices are better readings than those for the wet bulb temperature.

We have found that for the range of temperature over which these experiments were made, namely 90 to 110 deg. E. T., the dry bulb is a very poor index of the physiological reactions which will result. The wet bulb temperature is a better index though not so good as effective temperature. This is to be expected. If we look at this temperature range on a psychrometric chart, having all three scales of temperature, we find that the dry bulb lines far from coincide with the effective temperature lines while there is much less divergence between the wet bulb and the effective temperature lines. Along the 90 deg. effective temperature line 10 deg. D. B. corresponds to only 3 deg. in wet bulb temperature and along the 110 deg. effective temperature line, a change of 10 deg. D. B. represents only 0.7 deg. change in wet bulb temperature.

While wet bulb may be considered a fair index of physiological reactions and therefore of comfort over this temperature range, it does not follow that the same is true in the comfort region or other parts of the chart. The true comfort line (that is, the 64 deg. E. T. line) cuts 11.5 deg. W. B. for every 10 deg. D. B. This means that dry bulb in this region of the chart is a slightly better index than wet bulb while effective temperature is the only true index of physiological reactions and comfort.

No. 673

DETERMINATION OF THE COMFORT ZONE

By F. C. Houghten[1] (Member) and C. P. Yagloglou[2] (Non-Member)

Pittsburgh, Pa.

THE comfort zone has been the goal of much speculation, study, and experimentation during the past century. The heating and ventilating engineer, the physiologist and the physician were alike interested in the establishment of this zone of maximum human comfort.

Not many decades ago dry bulb was the only factor considered as an indication of a person's feeling of warmth. Seventy degrees dry bulb was assumed to be the ideal temperature, without reference to the wet bulb temperature or relative humidity.

The acceptance of humidity as a factor later led to an over-emphasis of its importance. The tendency in the past decade has been to entirely disregard the dry bulb temperature and accept the wet bulb temperature alone as determining our feeling of comfort. In accordance with this conception, 56 deg. wet bulb has received considerable support as the ideal comfort temperature.

The 56 deg. wet bulb line on the chart intersects the 70 deg. dry bulb line at 40 per cent relative humidity. True comfort is probably very near the intersection of these lines. Assuming this to be the case it was predicted in the presentation of the effective temperature chart that 63.5 deg. E. T. should be the comfort line. Before accepting this line however it was deemed advisable to verify it by further experiments at the Research Laboratory of the American Society of Heating and Ventilating Engineers. It was finally decided to make the determination of this line a part of very thorough study, the program being given below:

1. Determining the comfort zone for a representative group of people and the line of maximum comfort included in this zone. This is to be known as the comfort line. The boundaries of the comfort zone and the comfort line to be investigated independently at 30 and 60 per cent relative humidity in order to ascertain whether they coincide with the effective temperature lines. The lower and upper limits of the zone to be established by approaching them

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from both higher and lower temperatures so as to compensate for any possible effect of acclimatization.

2. Verification of the slope of the effective temperature lines in the comfort region by primary sense reactions of individuals wearing normal clothing.

3. Study of the slope of several effective temperature lines as affected by various types of clothing. An effective temperature line in the comfort zone, one considerably below and one considerably above this zone, are to be studied by six subjects, two dressed in tropical, two in summer and two in winter clothing.

FIG. 1. REPRESENTATIVE GROUP OF 130 PEOPLE WHO SERVED AS SUBJECTS IN DETERMINING THE COMFORT ZONE

4. Basal metabolism tests are to be made with at least three subjects, at three different relative humidities, on several different effective temperature lines between 55 and 85 deg. Skin temperatures of the subjects in the tests should also to be taken on the hands, face, and under the clothing.

5. From a consideration of the above, and other data which may be possible to obtain, an analysis should be made of the heat dissipated from the body by radiation, convection, evaporation, and other means.

The first two parts of this program have been completed and form the basis of this paper.

EFFECTIVE TEMPERATURE

The result of an investigation of the Relative feeling of warmth experienced by man in various combinations of temperature and humidity, the Laboratory recently published a chart showing a series of equal comfort or effective temperature lines. This series of lines, which is similar to the wet bulb and dry bulb series extends from the lower region of the chart representing conditions which feel cold, to the upper or hot region and forms a scale which is a true index of one's feeling of warmth. By giving numerical values to this scale it is possible to express by a single

[1] Determining Equal Comfort Lines, by F. C. Houghten and C. P. Yaglogiou, JOURNAL OF THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGRS., March, 1923, pp. 165-176.

number the relative feeling of warmth which one will experience for any combination of temperature and humidity. Because of the similarity between this series of lines and the wet and dry bulb temperature lines and further on account of the fact that one is accustomed to thinking of his feeling of warmth as being measured by temperature, it was thought logical that it should be called a scale of temperature. Since this new scale is a true index of all heat effects on the human body, both primary sense and physiological, it was called a scale of effective temperature. Its numerical value was logically fixed by the intersection of the lines with the saturation curve. That is, at saturation the effective temperature, wet bulb and dry bulb temperature, are always the same.

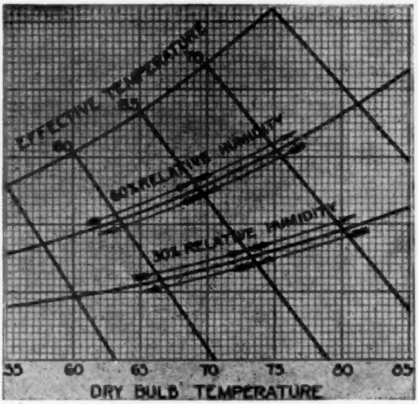


FIG. 2. CHART SHOWING RANGE AND CHANGES OF TEST CONDITIONS IN DETERMINING THE COMFORT ZONE

It is desirable at this time that a clear, concise and rigid definition of effective temperature be stated. This is not easy. Next to time, temperature itself is perhaps the most difficult physical measure to define. Perhaps the most accepted physical definition of temperature is: "The thermal condition of a body which determines the transfer of heat between it and other bodies." From this definition the temperature of air, that is its dry bulb temperature, determines the direction and magnitude of the flow of heat between the air and other bodies in contact with it. This is true so long as the body in contact with the air is dry. If the body is wet it will still be true so far as the transfer of heat by radiation and convection are concerned. The dry bulb temperature of the air will not however determine the total loss of heat from a wet body to the air.

The surface of the human body is not dry and the loss of heat from it to the air is therefore not determined by the dry bulb temperature but by the combined effect of dry bulb and relative humidity. Effective temperature is a single index which combines these two factors and we can say therefore: Effective temperature is the temperature and humidity condition, or the heat and moisture condition, of the air which determines the

transfer of heat between it and the human body. A more simple definition is: Effective temperature is an index of the degree of warmth which a person will experience for all combinations of temperature and humidity.

VERIFICATION OF THE EFFECTIVE TEMPERATURE LINES IN THE COMFORT ZONE

Two three-hour tests were made with eight subjects in the comfort zone to prove that in this region the effective temperature, and not the

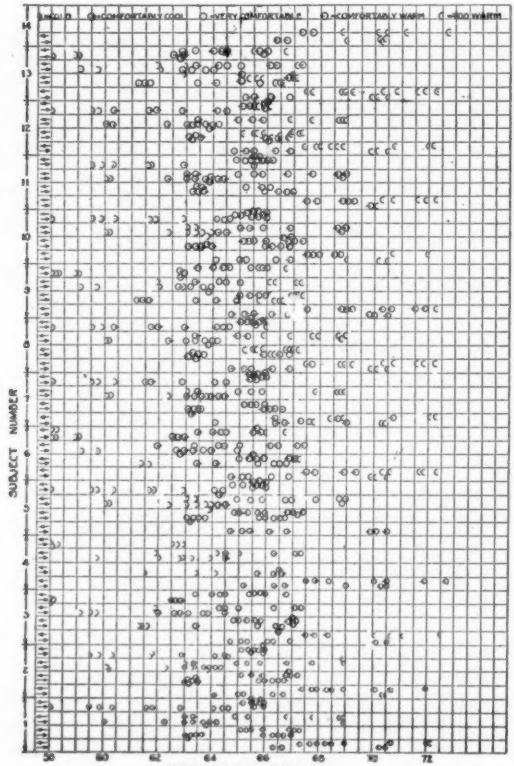


FIG. 3. DATA COLLECTED DURING THE EIGHT 2-HOUR PERIOD TESTS FOR DETERMINING THE COMFORT ZONE

wet bulb lines, are equal comfort lines for people normally dressed and remaining in these conditions for a considerable length of time. In the first test, on the 58 deg. wet bulb line, four subjects entered one of the rooms maintained at 58 deg. W. B. and 59 deg. D. B. (58.6 E. T.). The other four subjects entered the second maintained at 58 deg. W. B. and

80 deg. D. B. (68.8 E. T.). The eight subjects remained in the rooms as indicated, for 1½ hr. and then changed, those in the first room going into the second, and those in the second going into the first. Each subject kept an independent record of his own feelings during the three hours. Their various records are given in Table 1.

It will be seen that all subjects pronounced the first room (56 deg. W. B. 59 deg. D. B.) too cold. One judged the second room comfortable, one comfortable but warm enough to make him sleepy, one comfortable but preferred slightly warmer, while the fourth judged it too warm. All agreed definitely that the first was cooler than the second room.

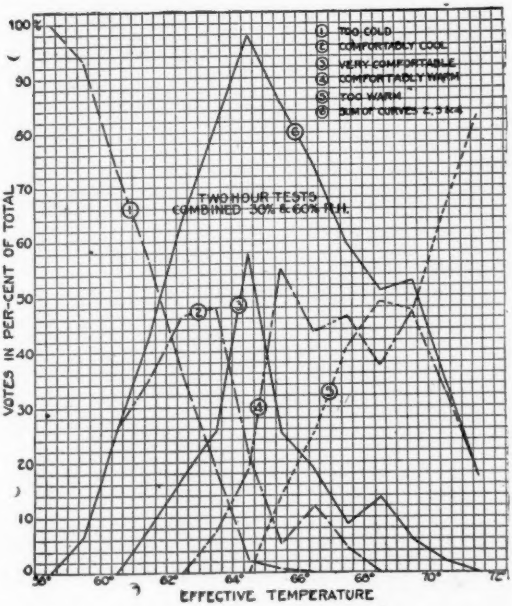


FIG. 4. CURVES SHOWING RELATION BETWEEN EFFECTIVE TEMPERATURE AND PERCENTAGE OF SUBJECTS RECORDING VARIOUS FEELINGS, DURING THE EIGHT 2-HOUR PERIOD TESTS AT 30% AND 60% RELATIVE HUMIDITIES

The second test was conducted in a similar way on the 65.0 deg. E. T. line, with 65.5 deg. D. B. and 64.5 deg. W. B. in the first room, and 74.0 deg. D. B. and 55.0 deg. W. B. in the second. The records of the various subjects in Table 2 show that the two conditions are very nearly the same.

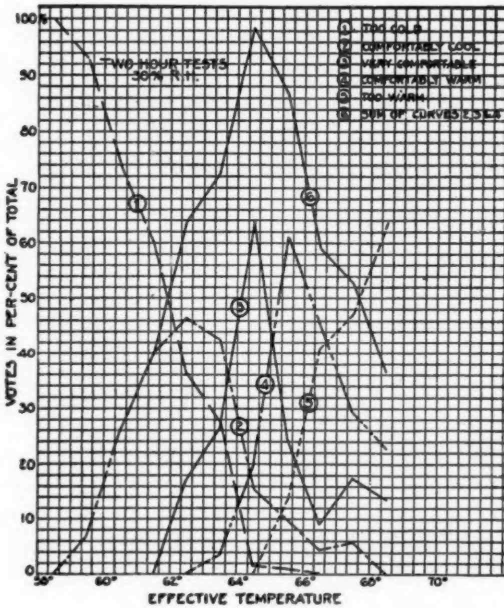
DETERMINATION OF THE COMFORT ZONE AND COMFORT LINE

Having definitely established the slope of the effective temperature lines in the comfort region the next problem was to locate the comfort zone and the comfort line within this zone. A large number of people, totaling 130 in all, participated as subjects in these tests. Great care was taken to get as nearly as possible a representative group of people. Individuals of both sexes, wearing widely different types of clothing and representing as many different occupations as possible, were chosen. Each was asked

to dress as he normally did in well ventilated living quarters. Fig. 1 shows part of the group of the employees of the U. S. Bureau of Mines and of the Research Laboratory who offered their services and to whom the Laboratory is greatly indebted.

Both psychrometric rooms of the Research Laboratory were used to accommodate the large number of subjects involved in the tests. The floors were covered with rugs. Comfortable chairs were provided, together with small tables for light activities of the subjects, such as card playing, reading and writing. Lights and room environment in general,

FIG. 5. CURVES SHOWING RELATION BETWEEN EFFECTIVE TEMPERATURE AND PERCENTAGE OF SUBJECTS RECORDING VARIOUS FEELINGS DURING THE EIGHT 2-HOUR PERIOD TESTS AT 30% RELATIVE HUMIDITY



were made cheerful. Outside air, washed and conditioned to the desired temperature and humidity was supplied to the rooms by the complete air conditioning equipment of the Research Laboratory.

METHODS OF CONDUCTING TESTS

A permanent record was made of the name, age, occupation, clothing and physical condition of each subject. A data sheet was furnished each subject upon entering the test room which required answers to the following questions at 15 min. intervals:

1. Is this condition comfortable or uncomfortable?
2. Do you desire any change?
3. If so, do you prefer warmer or cooler?

The room conditions were recorded at frequent intervals throughout the test. To accommodate the maximum number of subjects and to furnish data for a study of the problem in all its phases, three different series of tests were made and the results compared. For convenience of presentation they are grouped as follows:

1. Preliminary test.
2. Regular 2 hr. tests.
3. Short period tests.

PRELIMINARY TESTS

The object of the preliminary tests was to study the limits of the comfort zone and the effect of changing the room conditions. They were

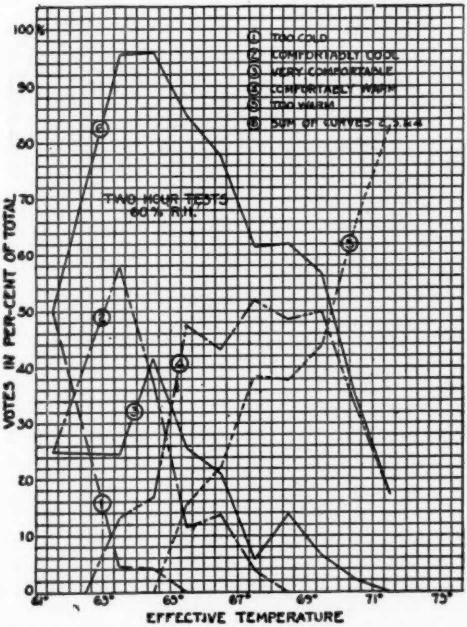


FIG. 6. CURVES SHOWING RELATION BETWEEN EFFECTIVE TEMPERATURE AND PERCENTAGE OF SUBJECTS RECORDING VARIOUS FEELINGS DURING THE EIGHT 2-HOUR PERIOD TESTS AT 60% RELATIVE HUMIDITY

made with subjects, the majority of whom were drawn from the personnel of the Research Laboratory and the U. S. Public Health Service. Some of these people were more or less familiar with the object of the tests and had had considerable experience in analyzing their own feelings. The subjects, twelve in number, remained in the test chamber for a period of 3

hr., from 9 to 12 in the morning and 2½ hr., from 1:00 to 3:30 in the afternoon.

The atmospheric conditions for the morning test varied from 60 to 71 deg. E. T., along the 50 per cent relative humidity line, stopping at every change three-quarters of an hour. In the afternoon the process was reversed. The temperature was stepped down from 71 to 59 deg. E. T. along the 50 per cent relative humidity line with practically the same number and period of stops.

REGULAR TWO-HOUR TESTS

The object of this series of tests was to obtain very thorough data on a large number of subjects within the limits of relative humidity met with in good ventilation practice.

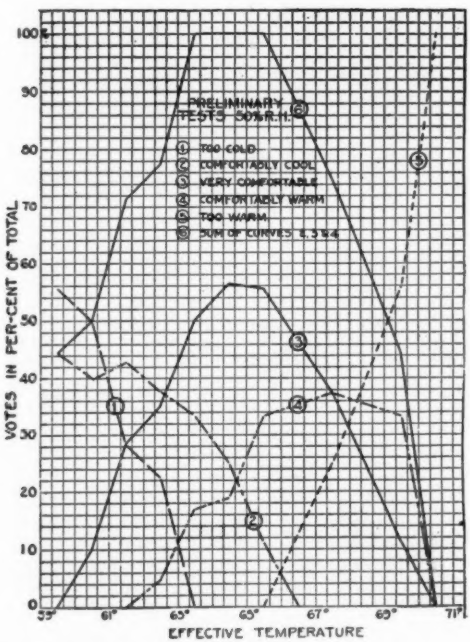


FIG. 7. CURVES SHOWING RELATION BETWEEN EFFECTIVE TEMPERATURE AND PERCENTAGE OF SUBJECTS RECORDING VARIOUS FEELINGS DURING THE PRELIMINARY TESTS

The number of people participating in this series of tests was limited to 14 by the capacity of the two rooms. It was considered inadvisable to crowd the subjects in the tests. Five women and nine men of various ages, types of clothing and occupations, as shown in Table 3, participated.

The same subjects took part in each of the eight 2-hr. tests, four at 30 per cent and four at 60 per cent relative humidity. To eliminate any effect of acclimatization to a temperature, when changing the conditions, the upper and lower limits of the zone were determined by approaching them from opposite directions. Two general methods were followed in conducting the tests. *First,* the test was started with an atmospheric condition outside the comfort zone for most people, either below or above, and the probable center of the zone was approached by steps of 1 or 2 deg. E. T. *Second,* the test was started with a condition within the zone for most subjects and then changed so as to move outward, either to a higher or lower temperature.

The range and direction of the change in temperature for the various tests are shown diagrammatically on the small psychrometric chart in Fig. 2.

SHORT PERIOD TESTS

It was found very difficult to obtain subjects who could be present at a large number of long tests. It was therefore decided to conduct a third series of short tests in which the rooms were maintained for two hours at various atmospheric conditions along the 30 per cent and 60 per cent relative humidity lines. During this period as many people as was possible to obtain entered the rooms for 15 min. or longer and filled out data sheets as shown above. One hundred individuals in all took part in these tests. Some participated in all of the tests of this series, while others took part in one or more tests. In this connection we are indebted to Prof. S. E. Dibble of Carnegie Institute of Technology, who was responsible for securing a large number of students as subjects.

DATA AND RESULTS

The method of analyzing the data obtained in the regular 2-hr. tests is shown in Fig. 3. Arrows at the left indicate the direction of the change in effective temperature for the various tests. The votes of the various subjects for the different effective temperatures are shown by the five characters given below. A close investigation will disclose that the majority of the comfort points lie between 63 and 66 deg. E. T.

Circle = very comfortable, no change in temperature desired.

Circle with dash pointing to the reader's right = comfortable, but if any change is to be made would prefer it slightly warmer. (We will refer to these as comfortably cool.)

Circle with dash pointing to the reader's left = comfortable, but if any change is to be made would prefer it slightly cooler. (We will refer to these as comfortably warm.)

Semicircle pointing to reader's right = uncomfortable; too cold.

Semicircle pointing to reader's left = uncomfortable; too warm.

The sum of all similar notation points within every degree effective temperature, expressed both in number and in per cent of the total number of points for the particular range, are tabulated in Table 4. Fig. 4 shows the results plotted against effective temperature. Starting at the lower

extreme boundary of the zone, that is at 58.5 deg. effective temperature, it will be noted that 100 per cent of the subjects are cold. As the effective temperature increases, the percentage of those feeling cold decreases, while that of the subjects being comfortable increases until at 63.5 deg., 25.6 per cent are very comfortable, 48 per cent comfortably cool, 7.2 per cent comfortably warm, and 19.2 per cent too cold. At 64.5 deg. the majority of the subjects, (57.8 per cent) are very comfortable as represented by the peak of Curve 3. As the temperature continues to rise the percentage of comfort votes drops, giving rise to Curve 5, which repre-

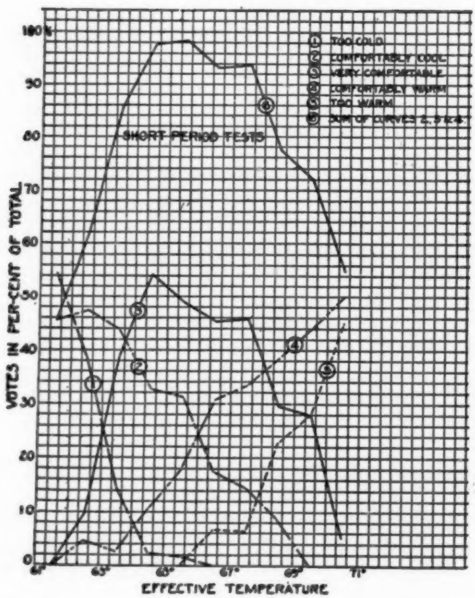


FIG. 8. CURVES SHOWING RELATION BETWEEN EFFECTIVE TEMPERATURE AND PERCENTAGE OF SUBJECTS RECORDING VARIOUS FEELINGS IN THE SHORT PERIOD TESTS

sents the percentage of the subjects feeling too warm from 64.5 to 71.5 deg. E. T. At the latter temperature 82.8 per cent of the subjects pronounced the condition too warm for comfort. By extrapolation it is found that 72.5 deg. E. T. is the upper extreme boundary of the comfort zone.

Curve 6 is the sum of Curves 2, 3 and 4 and represents the votes for total comfort in per cent. It will be noticed that it has the same characteristics as Curve 3, as is to be expected, and that both attain their maximum values at 64.5 deg. E. T.

Figs. 5 and 6 give respectively the results of the 2-hr. tests for 30 per cent and 60 per cent relative humidity, plotted against effective tempera-

ture. Maximum comfort again occurs at 64.5 deg. E. T. as represented by the peak of Curves 3 and 6 in both figures. This proves that comfort is a function of effective temperature, irrespective of the relative humidity, and that the slope of the effective temperature lines within the comfort region is correct.

The results of the preliminary tests were treated in a similar method as that employed in the previous series. Fig. 7 shows the votes in per cent of total plotted against effective temperature for 50 per cent relative humidity. The lower extreme boundary of the comfort zone is not well defined here, while the upper boundary falls at 70.5 deg. E. T., where the subjects unanimously pronounced the condition uncomfortably warm. At

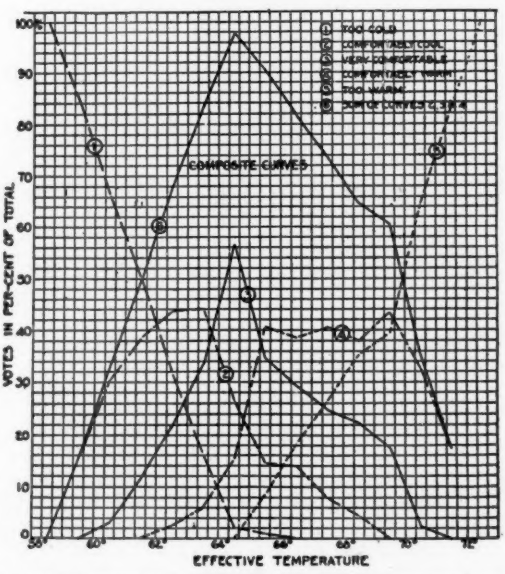


FIG. 9. COMPOSITE CURVES OF ALL THREE SERIES OF TESTS

59.5 deg., 55.5 per cent of the subjects were too cold and the remaining 44.5 per cent were comfortably cool. At 64.5 deg., 56.2 per cent of the subjects were very comfortable, 25 per cent comfortably cool and 18.8 per cent comfortably warm. The sum of these percentages is represented by the flat part of the total comfort Curve 6, extending over a range of two degrees from 63.5 to 65.5 deg. E. T. The general characteristic of all six curves are very similar to those of the previous series. Maximum comfort occurs at the same temperature of 64.5 deg. as represented by 100 per cent of the votes in Curve 6.

Similar plots for the short period series of tests are given in Fig. 8. The relative humidity during these tests varied between the two limits of 30 per cent and 60 per cent. It will be seen that the consensus of

opinion of the subjects points to the 64.5 deg. effective temperature as the one of maximum comfort, while the temperature for total comfort is not very sharply defined, but lies between 64.5 and 65.5 deg. The range of

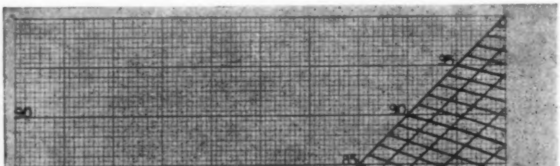


FIG. 10. EFFECTIVE TEMPERATURE CHART

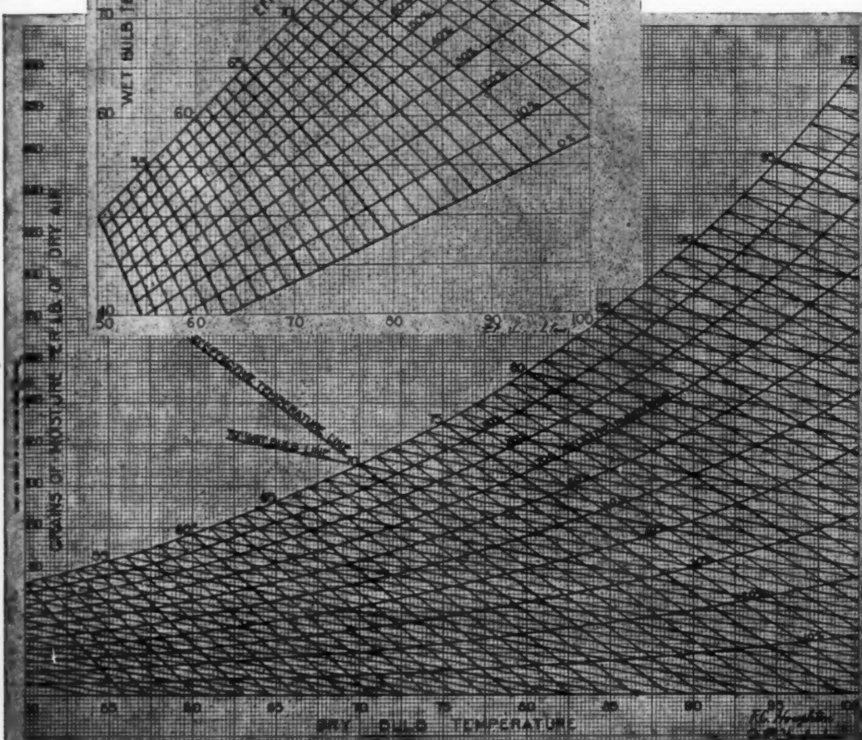


FIG. 11. PSYCHROMETRIC CHART SHOWING RELATION BETWEEN DRY BULB, WET BULB AND EFFECTIVE TEMPERATURE

temperature in these tests was not great enough to determine the upper and lower extreme boundaries of the comfort zone.

DISCUSSION AND CONCLUSIONS

The results of the various series of tests indicate that the general tendencies and characteristics of the curves are very similar among themselves and follow very closely the scale of effective temperature. They all point to the fact that 64.5 deg. E. T. is probably the true comfort line to be adopted as standard in heating and ventilating schools, theaters, residences and other places where mental work and light activities are carried on.

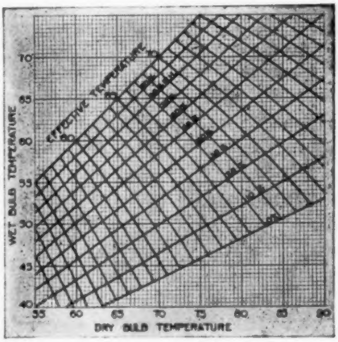


FIG. 12. PROPOSED CHART TO BE USED IN CONNECTION WITH THE SYNTHETIC AIR CHART

To better study the relation of effective temperature to comfort the results of the three series of tests are combined and given in Fig. 9. It will be noticed that at 58.5 deg. E. T., 100 per cent of the opinions, or in other words, all people present in the tests, agreed that the condition was too cold. This constitutes therefore the lower extreme boundary. The upper extreme boundary falls at 72.5 deg. E. T. obtained by extrapolation. At 64.5 deg. E. T., the percentage of those feeling too cold or too warm is practically zero while the percentage of the subjects being comfortable approaches 100 per cent. Of these 56.2 per cent are very comfortable, 26.3 per cent comfortably cool, 15.4 per cent comfortably warm and the remaining 2.1 per cent too cold.

It is remarkable to note in Fig. 9 that the increase in comfort and the decrease in discomfort from 58.5 to 64.5 deg. E. T. are straight line functions of effective temperature, as represented by the straight part of Curves 1 and 6. The same fact is apparent, although not so clearly pronounced in all plots of the three individual groups.

The rapid rise in comfort from 58.5 to 64.5 deg. E. T. followed by a comparatively gradual fall above 64.5 deg. indicates that the human element is more sensitive to lower temperatures than to high.

A careful examination of Fig. 3 discloses another interesting point, the effect of what may be called acclimatization, or becoming accustomed

to any temperature. It will be seen that if the temperature change is from lower to higher, the comfort zone and the comfort line shift to a slightly lower temperature and vice-versa. In other words, if the subjects dwell in the same temperature over prolonged periods of time, they become acclimatized to the condition. This does not affect the results, however, since tests were made with temperature changes in both directions.

The boundaries, width and center of the comfort zone for each of the 14 subjects in the two-hour tests are given in Table 3. These temperatures are taken from Fig. 3, and include only the zone in which the individual was always comfortable, comfortably cool or comfortably warm. Fifty-nine and five-tenths and 67 deg. E. T. are taken as the lower and upper boundaries of subject No. 1. While the lower boundary is very sharply defined for this subject, the upper boundary is not so sharply defined. Although he recorded some discomfort points as low as 67 deg., E. T. several higher comfort points were recorded. The width of the individual zones and their average as given in the table, is therefore smaller than that determined by the majority of votes.

The individual zones thus determined range in width from 2 to 8.5 deg. E. T. The average width is 5.4 deg. E. T. The center of the zones range from 62.5 to 68.7 deg. E. T. Sixty-five degrees effective temperature is the lowest upper boundary and the same temperature is also the highest lower boundary. There is, however, only one subject whose lower boundary is above 64 deg. E. T.

Attention is drawn to the two critical points of Curve 3 in Fig. 9, and the corresponding points of the individual groups of tests. Any small change in temperature between 63.5 and 65.5 deg. is accompanied by considerable variation in the percentage of votes for comfort. Outside these limits the change is rather gradual. Especially in Curves 3 of Figs. 4, 5 and 6 the two points are exceptionally well defined at 63.5 and 65.5 deg. E. T.

It is very fortunate that the human element is capable of adjusting itself to small variations of temperature from the comfort line, through the action of the thermostatic control within the body. The general tendency in this country is to overheat buildings with the very apparent results of reducing the efficiency and impairing the health of the occupants. Especially in industrial establishments where production depends chiefly on the efficiency of the workers, the problem of air conditioning should be carefully studied, to insure the best possible atmospheric conditions for the well being and maximum production of the workers.

It follows, therefore, that if a lower temperature is adapted as the standard of practice, the human element will soon become adapted to the new condition with the ultimate result of improving health and comfort and stimulating human activity.

LOCATION OF COMFORT ZONE

The location and width of the comfort zone can easily be determined from the data shown in Figs. 4 to 9 and in Table 3, if we agree on a

definition of the zone. If the zone is defined as including those temperatures over which 50 per cent of the people are comfortable, then it will be the distance between the two branches of Curve 6, Fig. 4, on the 50 per cent ordinate. This fixes the limits of the zone as 61.8 and 68.8 deg., or a width of 7 deg. The curves of the other series of tests give little variations from these values.

If the comfort line is defined as the line passing through the center of the zone then 65.3 deg. is that line. However, a consideration of the data in Fig. 3, Table 3 and the various curves show that this line does not pass through the greatest number of individual zones. It would seem advisable therefore to define the comfort line as that included in the largest number of individual zones. Accepting this definition, 64.5 deg. E. T. is the comfort line. It has already been shown that at this temperature 56 per cent of the subjects were very comfortable while 98 per cent were either very comfortable, comfortably cool or comfortably warm. In view of the fact that a round number should be accepted and also that a lower temperature, rather than a higher, is more conducive to health, 64 deg. E. T. is the most logical temperature to be adopted as the Comfort Line.

RELATION OF EFFECTIVE TEMPERATURE AND COMFORT LINE TO SYNTHETIC AIR CHART

One of the greatest recent developments in the art of heating and ventilation is the Synthetic Air Chart proposed by Dr. E. V. Hill. This chart provides a method of rating the percentage of perfection of atmospheric conditions. All conceivable factors are given a place in the chart with provisions for penalization for lack of perfection in any of these factors. The first thought of ventilation men upon considering effective temperature is the effect the latter will have on the working of the Synthetic Air Chart.

In keeping with the best information available the penalty for improper temperature and humidity has been based entirely upon wet bulb temperature. While this is not in accordance with the newly established effective temperature scale, it affects in no way the principle of the Synthetic Air Chart as a means of rating perfection in ventilation. All that is necessary is to adopt the effective temperature scale instead of the wet bulb scale as the basis for penalization. Effective temperature charts are given in Figs. 10 and 11. Fig. 10 is the standard psychrometric chart with the effective temperature scale. A more simple chart is given in Fig. 11, where the relation between dry bulb, wet bulb and effective temperature is more clearly shown. This chart can be readily used in cases where dew point and moisture content of the air are not desired.

Since effective temperature is not measured directly by any instrument but is a function of both wet and dry bulb temperatures, it is necessary in determining the penalty to record both wet and dry bulb readings in order to find the corresponding effective temperature from the chart. This is not a serious matter since it involves little additional trouble to take the dry bulb as well as the wet bulb reading. By printing the small

psychrometric chart of Fig. 12 on the Synthetic Air Chart, the effective temperature can very easily be obtained. The penalty will then be determined by the difference between the effective temperature of the condition and the comfort line (64 deg. E. T.).

ACKNOWLEDGMENTS

The writers are indebted to W. H. Carrier and Dr. E. V. Hill for their valuable suggestions and kindly criticisms. Acknowledgment is also due to those employees of the U. S. Bureau of Mines, U. S. Public Health Service, the A. S. H.-V. E. Research Laboratory, and to those students of the Carnegie Institute of Technology who offered their services in judging the conditions of tests.

TABLE 1. TEST ON 58 DEG. WET BULB LINE

Condition in First Chamber—59 deg. D. B., 58 deg. W. B., 58.6 deg. E. T.
Condition in Second Chamber—80 deg. D. B., 58 deg. W. B., 68.8 deg. E. T.

SUBJECT: W. Fulton
Laboratory Aid, U. S. Bureau of Mines

	2nd Chamber		*1st Chamber*
11:20	Entered second chamber passing through first chamber.	1.00	Changed from second to first chamber.
11:20	Second chamber feels much more comfortable than first chamber. First chamber much colder than second chamber.	1:05	First chamber much colder than second chamber and not the least bit comfortable.
12:15	Second chamber feels very comfortable, in fact, ideal.	1:45	I feel chilly.
12:50	At the end of the hour and one-half this room feels very comfortable.	2:30	Still chilly.

SUBJECT: J. V. Boehm
Night Academy Student: Employed as subject in Tests

	2nd Chamber		*1st Chamber*
11:20	Entered second chamber passing through first chamber.	1:00	Changed from second to first chamber.
11:20	Room one cold compared to room two.	1:15	Too cool.
11:25	Just comfortable.	1:48	Colder.
12:10	Would like about three degrees warmer.	2:30	Cold: run along backbone.
12:50	Comfortable.		

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SUBJECT: C. A. Herbert
Night Academy Student: Employed as subject in Tests

	2nd Chamber		*1st Chamber*
11:20	Entered second chamber passing through first chamber.	1:00	Changed from second to first chamber.
11:20	Room two warmer than room one. Room one felt cold passing through.	1:00	Room one much colder than room two.
11:45	Room two could be cooler for comfort.	1:10	Too cold for comfort.
12:00	It seems a little cooler now.	1:20	Colder.
12:10	It feels just right now.	1:45	Feel a little chilly.
12:30	Comfortable.	2:10	More chilly.
12:50	The same.	2:25	Still Chilly.

SUBJECT: F. C. Houghten

2nd Chamber

11:20 Entered second chamber passing through first chamber.
11:20 Chamber two decidedly warmer than chamber one.
11:45 Feels too warm for comfort.
12:50 Same as 11:45.

1st Chamber

1:00 Changed from second to first chamber.
1:00 Feels refreshing.
1:30 Decidedly cool.
2:20 Too cool for comfort.

SUBJECT: Mr. Riley
Laboratory Aid, U. S. Bureau of Mines

1st Chamber

11:20 Entered first chamber.
11:35 Noticeably cool.
12:00 Becoming chilly.
12:15 It is too cool for comfort.
12:50 The same.

2nd Chamber

1:00 Changed from first to second chamber.
1:00 Second chamber feels agreeably warmer upon entering.
1:20 Am more comfortable now.
2:00 Am comfortable, but would feel more so if warmer.
2:40 Room is comfortable, but prefer it warmer.

TABLE I. TEST ON 58 DEG. WET BULB LINE—*Continued*

SUBJECT: R. L. Lincoln
Laboratory Assistant

1st Chamber

11:20 Entered first chamber.
11:20 Slightly cool but refreshing.
12:00 Same feeling.
12:15 Cool.
12:50 Slightly chilly.

2nd Chamber

1:00 Changed from first to second chamber.
1:00 Feel warmer in this room.
1:30 Very comfortable.
2:00 Slightly too warm for comfort.
2:40 Still comfortable. No desire to study. Slightly drowsy.

SUBJECT: Mr. Guth
Junior Chemist, Bureau of Mines

1st Chamber

11:20 Entered first chamber.
11:20 Room one feels cooi and fresh.
11:40 Same as at 11:20.
11:45 Not too cool for comfort.
11:50 Continue to feel the same.
12:25 Begin to feel cooler.
12:30 Cooler still.
12:50 Chilly.

2nd Chamber

1:00 Changed from first to second chamber.
1:00 Second room feels considerably warmer than first.
1:05 Feel very comfortable.
1:15 Slightly drowsy.
1:40 Same as at 1:15.
2:00 Still drowsy.
2:40 Still drowsy. Room remained comfortable throughout.

SUBJECT: Dr. McConnell

1st Chamber

11:20 Entered first chamber. Room one feels decidedly cool for comfort throughout the test.

2nd Chamber

1:00 Changed from first to second chamber. Room two feels decidedly warmer than room one and comfortable.

TABLE 2. TEST ON 65 DEG. EFFECTIVE TEMPERATURE LINE

Condition in First Chamber—65.5 deg. D. B., 64.5 deg. W. B., 65 deg. E. T.
Condition in Second Chamber—74 deg. D. B., 55 deg. W. B., 65 deg. E. T.

SUBJECT: W. Fulton
Laboratory Aid, U. S. Bureau of Mines

2nd Chamber

11:30 Entered second chamber passing through first chamber.
11:35 Did not feel there was any difference between the two rooms as to heat, but room two was noticeably dry.
12:15 This room could be a little warmer to be comfortable. Same feeling throughout.

1st Chamber

1:30 Changed from second chamber to first chamber.
1:35 The two rooms to me are the same as far as heat, but both could be a little warmer for comfort.
2:15 Same feeling throughout.

SUBJECT: J. V. Boehm
Night Academy Student: Employed as subject in Tests

2nd Chamber

11:30 Entered second chamber passing through first chamber.
11:30 Both rooms same and comfortable.
12:15 Still comfortable.
12:50 Comfortable throughout.

1st Chamber

1:30 Changed from second chamber to first chamber.
1:35 Could not tell that there was any difference in temperature.
2:15 Same before leaving.

SUBJECT: C. A. Herbert
Night Academy Student: Employed as subject in Tests

2nd Chamber

11:30 Entered second chamber passing through first chamber.
11:30 Very little difference between room one and room two.
12:15 Both rooms cool.
12:50 Continued cool until leaving.

1st Chamber

1:30 Changed from second to first chamber.
1:35 A little cooler than room two. Not so comfortable.
2:15 Feel a little chilly throughout the test.

TABLE 2. TEST ON 65 DEG. EFFECTIVE TEMPERATURE LINE—*Continued*

SUBJECT: Mr. F. C. Houghten

2nd Chamber

11:30 Entered second chamber passing through first chamber.
11:30 The two rooms are practically the same.
12:30 Room No. 2 very comfortable.

1st Chamber

1:30 Changed from second chamber to first chamber.
1:30 Could not notice any difference in sense of warmth of the two rooms.
2:15 Comfortable throughout.

SUBJECT: R. Otter
Laboratory Assistant

1st Chamber

11:30 Entered first chamber.
12:00 General feeling for comfort.
1:00 A growing feeling of cold evident.
1:25 Feeling of cold, almost amounting to a chill.

2nd Chamber

1:30 Changed from first to second chamber.
1:35 The first impression on entering room two was one of warmth, but this gradually changed to one of neither warmth nor cold.
2:00 A slight feeling of cold.
2:30 No noticeable change in the feeling of cold.

SUBJECT: Louis Ebin
Laboratory Assistant

1st Chamber

11:30 Entered first chamber.

12:00 Room one feels slightly cool for comfort. Same feeling throughout.

2nd Chamber

1:30 Changed from first to second chamber.

1:35 First impression on going from room one to room two is that the latter is considerably warmer.

1:45 It now seems cooler than at 1:35, but more comfortable than in room one.

2:00 Feel more comfortable in this room. I think this is greatly due to being more absorbed in other things. Same feeling throughout test.

SUBJECT: J. Brown
Laboratory Assistant

1st Chamber

11:30 Entered first chamber.

11:30 First room comfortable. Same throughout.

2nd Chamber

1:30 Changed from first to second chamber.

1:30 Second room feels slightly warmer than first room.

1:50 Second room feels cooler than at 1:30.

2:25 Second room feels very comfortable.

SUBJECT: Dr. McConnell

1st Chamber

11:30 Entered first chamber. A growing feeling of cold is apparent in this room. A feeling of being chilly was experienced before leaving room.

2nd Chamber

1:30 Changed from first to second chamber. First impression upon entering this room was that it was slightly warmer than room one. This may be due to more cheerful atmosphere. A growing feeling of coolness equaling that in the first room developed. Upon leaving the room I am of the opinion that the two rooms are the same as far as warmth is concerned.

TABLE 3. DATA OF SUBJECTS IN TWO HOUR TESTS

Subject No.	Sex	Age	Weight	Occupation	Clothing	Comfort Zone Boundaries in Terms of Effective Temperature: Lower fahr.	Upper fahr.	Width fahr.	Center fahr.
1	Male	19	163	Farmer	Heavy	59.5	67.0	7.5	63.2
2	Female	18	123	Secretary	Extra light	64.0	69.0	5.0	66.5
3	Male	54	135	Laborer	Medium	62.0	70.0	8.0	66.0
4	Male	22	142	Lab. Assistant	Heavy	65.0	72.5	7.5	68.7
5	Male	20	140	Student	Heavy	64.0	70.5	6.5	67.2
6	Female	43	132	Draftsman	Light	61.5	66.0	4.5	63.7
7	Male	21	157	Operating Engr.	Light	64.0	67.5	3.5	65.7
8	Male	22	130	Mechanic	Medium	60.0	65.0	5.0	62.5
9	Female	20	94	Clerk	Extra light	63.0	65.5	2.5	64.3
10	Female	18	123	Housekeeper	Light	64.0	69.0	5.0	66.5
11	Male	29	170	Research Engr.	Heavy	61.5	70.0	8.5	65.7
12	Male	19	130	Lab. Assistant	Medium	60.0	65.0	5.0	62.5
13	Female	23	100	Stenographer	Light	63.0	65.0	2.0	64.0
14	Male	22	135	Stock Clerk	Medium	......	68.0	......	.....
Average						62.4	67.8	5.4	65.1

TABLE 4. RESULTS OF TWO HOUR TESTS

	Summation of Similar Comfort Votes						Votes in Per Cent of Total					
Effective Temperature	Cold	Comfortably cool	Very comfortable	Comfortably warm	Too warm	Total	Cold	Comfortably cool	Very comfortable	Comfortably warm	Too warm	Total comfort = 2+3+4
	1	2	3	4	5	6	1	2	3	4	5	6
58° to 59°	26	0	0	0	0	26	100.0	0.0	0.0	0.0	0.0	0.0
59° to 60°	27	2	0	0	0	29	93.1	6.9	0.0	0.0	0.0	6.9
60° to 61°	17	6	0	0	0	23	73.9	26.1	0.0	0.0	0.0	26.1
61° to 62°	21	13	3	0	0	37	56.7	35.2	8.1	0.0	0.0	43.3
62° to 63°	21	27	10	0	0	58	36.2	46.5	17.3	0.0	0.0	63.8
63° to 64°	24	60	32	9	0	125	19.2	48.0	25.6	7.2	0.0	80.8
64° to 65°	2	20	55	18	0	95	2.1	21.1	57.8	19.0	0.0	97.9
65° to 66°	1	9	44	97	25	176	0.6	5.1	25.1	55.0	14.2	85.2
66° to 67°	0	14	22	51	30	117	0.0	12.0	18.8	43.5	25.7	74.3
67° to 68°	0	3	6	32	28	69	0.0	4.4	8.7	46.3	40.6	59.4
68° to 69°	0	0	7	19	25	51	0.0	0.0	13.7	37.3	49.0	51.0
69° to 70°	0	0	1	8	8	17	0.0	0.0	5.9	47.0	47.1	52.9
70° to 71°	0	0	1	16	32	49	0.0	0.0	2.0	32.5	65.5	34.5
71° to 72°	0	0	0	5	24	29	0.0	0.0	0.0	17.2	82.8	17.2

DISCUSSION

F. PAUL ANDERSON: I should like to ask Mr. Houghten to explain that equivalent temperature line.

F. C. HOUGHTEN: You can probably all remember the time when nothing but dry bulb temperature was thought of in practice. With the acceptance of the idea that humidity is an important factor in determining comfort we changed our mind. We know that if we have a high humidity and say 90 deg. we may be warmer than we would be at 100 deg. with a lower humidity. If that is true there must necessarily exist a certain humidity at 100 deg. dry bulb temperature which gives us the same feeling of warmth as 90 deg. with 100 per cent humidity. There must be a large combination of temperatures and humidities that give the same sense of warmth.

Such combinations of temperature and humidity which give us the same sense of warmth when plotted on a psychrometric chart give lines of equal sense of warmth or, equal comfort in so far as our feeling of warmth is concerned and those lines have for some time previous to their establishment been designated by the term equal comfort lines. The particular line passing through the temperature which we desire for comfort in heating our homes and dwelling places has been called the comfort line.

After establishing these lines, there was considerable criticism to calling them comfort lines, because at high temperatures, there is no comfort. We then came to the conclusion that we might better call them effective temperature lines. The particular effective temperature line which passes through those conditions of temperature and humidity which we accept as being desirable for heating our dwellings and public places will be called

the comfort line, or the combination of lines, since there is a certain range of temperatures over which we are comfortable, the comfort zone.

F. PAUL ANDERSON: I think it would be desirable if this Society would recognize for all time what we mean by equivalent temperatures. Equivalent temperature isn't anything that exists. It is an idealism that shows the relationship between dry bulb and wet bulb temperature, and the points coming on these lines of equivalent temperature represent an ideal condition. In other words, any point on that line would represent the same degree of comfort. Equivalent temperature lines are made up of a series of points which all represent practically the same amount of heat or temperature, not heat but heat intensity, one representing just exactly the same value as the other.

L. A. HARDING: Will you give a definition for effective temperature?

F. C. HOUGHTEN: Effective temperature is a scale which determines all our feelings of warmth and physiological reactions.

E. VERNON HILL: One that I have used (possibly the Research Laboratory does not subscribe to it) is giving off the normal amount of heat without undue exercise. That is, if you give off the amount of heat you are generating without calling upon your system to shut it down or increase it, then you are comfortable. As you go down in dry bulb temperature and decrease the humidity you give off less heat by convection and more heat by evaporation, but the total amount of heat given off will be the amount of heat to keep the body comfortable.

A MEMBER: Why not define effective temperature as that combination of temperature and humidity that will permit an equal heat loss or heat addition to the body?

W. L. FLEISHER: I should like to ask whether measurements were taken in still air and what effect that would have.

F. C. HOUGHTEN: This work was all done in still air. We are continuing work now in moving air.

W. L. FLEISHER: That might change the entire results because in a good many school rooms you have moving air.

F. C. HOUGHTEN: If you have moving air at ordinary temperature you will require slightly higher temperature for the same degree of comfort.

E. VERNON HILL: During all the discussion of this very important work I have heard no mention made of the pioneer experiments of Prof. John W. Sheppard who started his work, I believe, in 1913 in the Teachers' Normal College in Chicago. He is responsible for introducing the term *comfort zone,* and he was the first, so far as I know, to point out the relationship between temperature and humidity in its bearing on comfort. I think his name and his work should be mentioned by us in carrying on these experiments and it should go into the records.

J. I. LYLE: I don't believe we are ever going to use effective temperature in our specifications as to temperature at which a building is to be heated; we always do that in terms of wet bulb and probably relative humidity, but I might point out that to some of us the work already is of a great deal of value. For instance, I might call attention to the fact that in a good many textile mills where it is necessary on account of the work to use very high humidities we have found the building that is held with a high humidity is very much more comfortable for the operatives than a similar room at a somewhat higher temperature and lower humidity.

In a case the other day I think we figured an outside temperature of 95 deg. with normal humidity of about 30 per cent and by raising the humidity to 80 per cent we reduced the effective temperature as shown by this chart 5 deg. Five deg. cooler although we had added the moisture and, of course, from the millman's standpoint that is very important because his work runs better at a high humidity, and if we have reduced the effective temperature 5 deg., it is very much better for his employees.

JOHN G. SHODRON: I have tried to make an analysis of some advance information that I received last week on this 64½ deg. effective temperature line and have probably gone into the field a little bit beyond the paper.

In man the process of digestion and resorption, or the transformation of food into substances required for maintenance, growth, and energy values for activities, manifests itself in two ways—thermogenesis, or the development of heat inside the body and thermolysis, or the loss of heat from the body.

Men of different activities do not digest the same quantity of any given food; on the other hand, they maintain an internal body temperature which is substantially constant and varies slightly with external conditions and varies distinctly during the day. The daily variation has been measured by many observers and shows individual peculiarities which depend largely upon the time of meals, manner of living and type of food. In general, the body is at its lowest temperature early in the morning, rising slowly during the day, reaches its maximum early in the evening and falls again during the night. The difference between the early morning and late afternoon temperatures may amount to as high as 2 deg. fahr. Muscular activity and food appear to be the main factors that are responsible for the rise in temperature during the day.

The prime factors in determining man's state of comfort, it appears, are the dry and wet bulb temperatures of the surrounding objects and wall surfaces. Certain combinations of these factors will produce a state of maximum comfort while other combinations will cause a rise in body temperature or compel the body to oxidize body tissue in order to maintain its normal body temperature.

Combinations of dry and wet bulb temperatures, rate of air motion and radiation constants at which the outflow of heat just balances the production of internal heat, are factors of physiological and economic significance because the quantity of heat a man produces varies with

the amount and kind of work he performs and the nutrient content of the food he consumes.

Obviously, no single factor or combination of two or three factors can be considered as producing lines of equal comfort or effective temperatures without considering and establishing the factors causing the transfer of heat from the body by radiation, the quantity and kind of work and the general climatological conditions.

The quantity of heat which a surface radiates is independent of the temperature of the air or the distance between it and the surface it radiates to, but is dependent upon the character and the difference of the fourth powers of the absolute temperature of the surfaces in question. Applying the Stefan-Boltzman law to condition of man and his surroundings during the heating season, it will be at once seen that the heat transfer by radiation becomes a very important factor in the determining of comfort zones and lines of effective temperature.

Thus a change of wall surface temperature of 10 deg. may cause as much as 50 per cent increase in the transfer of heat by radiation from the body. Further, since the comfort of the human body depends upon its rate of heat loss, the maximum comfort must be found at a point where the quantity of heat given off just balances the amount of heat produced and this critical effective temperature zone must have a constant heat absorbing power.

The point is that this effective temperature line is a line at which the body gives off an equal amount of heat for the entire body. If a line of a higher degree is taken, then the body being dependent for its comfort on the amount of heat it gives off cannot give off the approximately 411 heat units per hour, and is caused a certain amount of discomfort until it has regulated itself so that it can work under a condition to give off less heat. Again, if the degree of the effective temperature line is under 64½, or whatever is set for maximum comfort, then the body must produce more heat and as a maximum comfort is dependent upon its rate of heat loss, the quantity of heat given off for maximum comfort must just balance that amount of heat which is produced at that particular temperature.

F. PAUL ANDERSON: In the discussion, such as the determination of the equivalent temperature lines, there are an infinite number of variables. We are trying to get away from these variables and determine what the law is by a process with which you gentlemen no doubt are familiar. The first effort was to determine a law of cooling. The next effort was to take a few selected subjects and carry them through a wide series of investigations. The next step was to take a large number of individuals and the results that were presented yesterday represent what has been accomplished, dealing with a relatively large number of subjects. This whole question of the normal human being is something that perhaps man will never be able to put his finger on. There is such a variation in the way people dress and the way people eat that it will be impossible to lay down a law that affects every person, but what we are attempting to do is to take a large range of subjects, as Dr. Hill expressed it, we take

people dressed from the flapper to the hack driver. Well, that included pretty nearly everything we could think of in the way of a subject.

I want to call your attention to a program that has been outlined, because it has not been mentioned before. There is a committee known as the Committee on Atmosphere and Man, the chairman of which is Ellsworth Huntington, the author of *Civilization and Climate,* and one of the eminent authors of this country, a man who has been thinking about this subject for years. This committee met the other day with a National Industrial Conference Board in New York for the purpose of outlining a series of experiments on very large numbers of humans at work. The plan now is to take about ten of the factories of this country and make a study of comfort and then health as compared with the productive value of these individuals. When that is accomplished perhaps we will have something that would represent more nearly the average condition, the ideal condition under which man should live. I agree with the speaker that all these factors should be considered, whether the tests are made in the afternoon or in the morning; the amount of food that is eaten at lunch time and the character of it, of course, would affect this whole question, but in a large number of subjects you find some of them do not eat lunch, others drink milk, some eat beef and where a large number of subjects are taken, even in morning and afternoon, the chances are you reach a very satisfactory average. Of course, we are not going to take care of the fellow who goes and eats such a dinner as we had last night and expect him to feel comfortable.

No. 674

EFFECT OF PRESSURE ON HOT WATER CIRCULATION

By Frank B. Rowley[1], Minneapolis, Minn.

Member

IN the gravity hot water heating system, the question often arises as to whether pressure in the system affects the circulation or whether circulation is due to temperature alone. Several systems have been designed in which the pressure has been increased and a few in which it has been decreased from that obtained in the open tank system. In both systems the claim is made that the circulation is increased. In performing tests on a certain heating system in the Experimental Engineering Laboratories at the University of Minnesota, it was necessary to determine the effect of pressure on the velocity in the system. It is with the thought that the results obtained may be of some interest to heating engineers that this article is written.

In performing the tests four radiators were connected to an Ideal Arco radiator boiler as shown in Fig. 1. The system was arranged with an expansion tank which might either be closed off by a valve just below the tank or left open. The boiler was fired with a gas heater, the gas first being passed through a pressure regulating device in order to insure uniform supply to the boiler. The boiler was set in the basement with two supply and return lines leading from it. Each line supplied one radiator on the first and one on the second floor. It was provided with water supply and discharge lines such that the pressure in the system could be varied from atmospheric pressure at the boiler to 30 or more pounds gage. At the top of radiator No. 4 an inverted glass U-tube was connected in order to determine whether or not air collected in the system during low pressure runs.

The circulation of water in the system was determined by four meters, details of which are shown in Fig. 2. These meters consisted of a very light brass runner or paddle wheel, which was supported by pivot bearings and placed in a heavy glass tube. The tube was placed between two flanged pipes provided with proper gaskets to prevent leakage. These meters were calibrated, two of them placed in the main flow lines from the boiler and two in the branch lines leading to the second floor radiators

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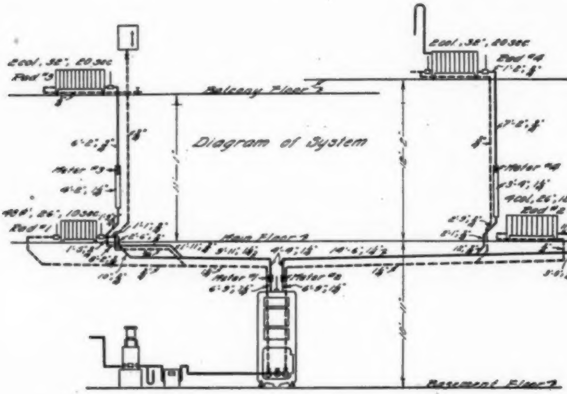

FIG. 1. ARRANGEMENT OF TEST APPARATUS

as shown in Fig. 1. The flow was determined by taking the time with a stop watch for a certain number of revolutions of the paddle wheel.

Three series of tests were made: First, to determine the efficiencies and circulation in the system under different water pressures, the other conditions remaining constant; second, to determine the rate of circulation through the system as the boiler was brought up from a cold condition; third, to determine the effect on circulation of changing the pressure when in operation.

In the first series three tests were made—with 30 lb., 15 lb. and 0 lb. gage pressure, respectively, on the boiler. In making the tests the boiler was first brought up to normal operating condition and the following readings were taken every five minutes throughout the test: pressure, volume, and temperature of gas burned, boiler pressure, temperature inlet and outlet of radiators Nos. 1 and 4, temperatures inlet and outlet of boiler, revolutions per minute meters Nos. 1, 2, 3, and 4, air temperatures at each floor, and flue gas temperature. Barometer reading and flue gas analysis was taken once for each test, the flue gas samples being collected throughout the run.

The average results of these tests are shown in Table 1, an examination of which shows that while the water circulation is nearly the same in

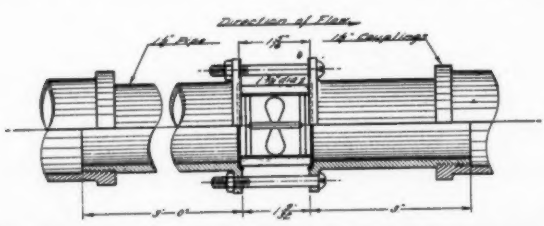

FIG. 2. DETAIL OF METER FOR DETERMINING CIRCULATION

TABLE 1. AVERAGE RESULTS OF TESTS

No. of Run	Length of Run	Avg. Gage Press.	Cu. ft. Gas Per Hr.	B.T.U. Per Cu. ft.	Total B.T.U. Sup.	Total hourly circulation: Meter No. 1	Meter No. 2	Total	Average Boiler Temp.: In	Out	Diff.	Total B.T.U. Per Hr.	Flue Gas: Temp. F°	% CO_2	% CO	% O_2
1	1	30"	7290	600	43740	592	563	1155	138.8	166.0	27.2	31416		9.2	0	7.0
2	1	15"	7265	600	43590	595	560	1155	138.7	166.3	27.6	31900	557	12.5	.05	5.8
3	1	0	7290	600	43740	588	552	1140	133.7	166.1	28.4	32350	588	12.8	0.8	5.0

No. of Run	Eff. of Boiler	Hourly water circulation in radiator in pounds: Meter No. 3	Rad. No. 3	Rad. No. 1	Meter No. 4	Rad. No. 4	Rad. No. 2	Radiator No. 1 Avg. Temp. F°: In	Out	Diff.	B.T.U. Rad. Per Hr.	Radiator No. 4 Avg. Temp. F°: In	Out	Diff.	B.T.U. Rad. Per Hr.
1	66.0	349	349	243	340	340	223	159.4	146.8	12.6	3062	156.5	143.1	13.2	4488
2	68.5	322	322	273	342	342	218	159.0	147.4	11.6	3167	155.9	142.8	13.1	4480
3	68.0	313	313	275	323	323	229	154.8	142.3	12.1	3327	153.1	140.5	12.6	4063

all tests, it is slightly less in Test 3 at 0 gage pressure. The temperature drop between the riser and return lines is also slightly greater in this test indicating that there is a slight decrease in circulation, if the pressure is reduced below the open tank system. This difference, however, is not great. In Tests 1 and 2 the circulation and the temperature drop are practically the same which would indicate no advantage in increasing the pressure over that of an open tank system.

The results of Tests 4, 5 and 6 of series No. 2 are best shown in the curve sheets Figs. 3 and 4. This series was made to determine the action of the system when being brought up to a working temperature from a cold condition. In each test the fire was turned on in a cold boiler and all readings were taken each five minutes as before for a period of one and one-half hours. The pressures used in Tests 4, 5, and 6 were 30 lb., 15 lb. and 0 lb., respectively, at the boiler.

The curve of Fig. 3 shows the outlet temperature for each of the tests and the difference between outlet and inlet water temperatures. The outlet temperature reached approximately 140 deg. at the end of the run. Therefore, the system did not reach the temperatures at which series No. 1 was run.

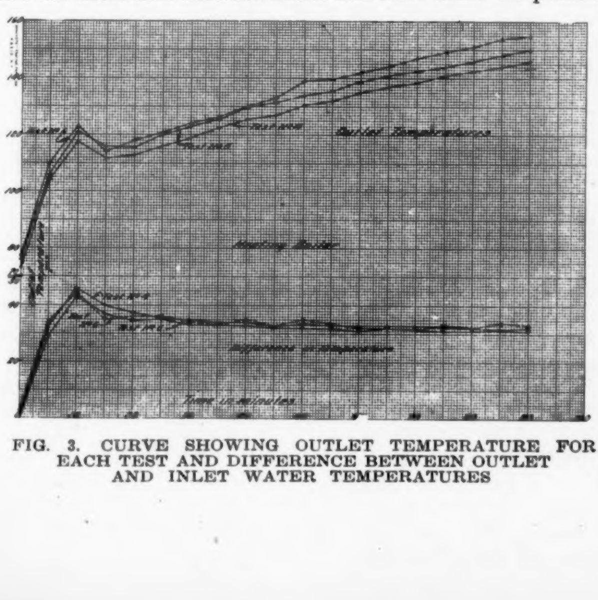

FIG. 3. CURVE SHOWING OUTLET TEMPERATURE FOR EACH TEST AND DIFFERENCE BETWEEN OUTLET AND INLET WATER TEMPERATURES

The curves of Fig. 4 show the pounds of water circulated through

the boiler. The results of this series show a slight difference in favor of the 30 lb. pressure. A set of readings taken at the end of a two hour period showed that the difference in circulation was decreasing. In Test 4, 126.4 cu. ft. of gas was burned and in Test 5, 126.3 and in Test 6, 126.6 cu. ft. This series would indicate that the difference in circulation is small but is slightly in favor of pressures.

The third series of tests was made to determine the effect on circulation, of pressure variation during the operation of the system. In this

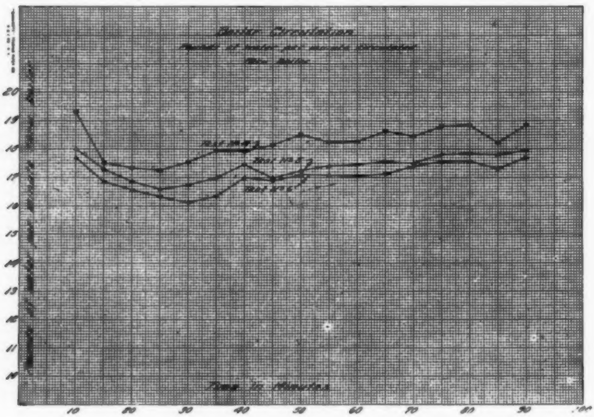

FIG. 4. CURVE SHOWING POUNDS OF WATER CIRCULATED THROUGH BOILER

series the pressure was changed from 0 lb. to 30 lb. and vice versa by admitting water into or drawing water from the boiler. The change in pressure disturbed the system and in general the circulation was somewhat increased by lowering the pressure and decreased by raising it. The decrease would be expected as some cold water was admitted to the system to raise the pressure. As soon as the boiler had recovered its normal temperature, the circulation returned to normal, and was about the same for the low as for the high pressures.

All of the tests were made to determine the effect of pressure on circulation, the heat supplied and the temperature being kept, as nearly as possible, the same for all tests of each series, no advantage being taken of the fact that with one system a higher temperature might be maintained than in another. In general, it appears that the effect of pressure on circulation is very slight, any difference being in favor of the pressure system, that is, a pressure at least equal to the open tank system. While the pressures in these experiments were brought to atmospheric at the boiler-line, therefore below atmospheric at the top of the system. There is a lower limit to which the pressures can be carried due to the vapor pressure which at high water temperatures would cause the water to drop below the top of the system and stop the circulation. Another difficulty in trying to operate a system on any pressure below the open tank pressure is that of keeping the system free from air which will collect at the top of the system and stop circulation.

No. 675

HEAT AND HUMIDITY CONTROL IN BUILDINGS

By E. T. Murphy, Philadelphia, Pa.

Member

WEATHER has been a subject of absorbing interest from the beginning of life on this earth. It has been a factor in the distribution of population, color and occupation and even more so in the distribution of agriculture, civilization and industry. Just so long as man accepted weather as inevitable, he made little or no progress. When he housed himself in some protective coating, he began literally to make his own weather and then adding fire, he was able to prophesy with reasonable accuracy, what at least his personal body weather would be next moon.

While climate then had less to do with the distribution of people it is astonishing how recent is the development of the idea of housing industry in its own climate. For hundreds of years, the industry had gone to the favorable climate, enriching some and keeping others in poverty.

For example, 15 years ago the writer visited many of the mill towns of the South right in the cotton fields located miles from a railroad in order to obtain cheap water power. The mills could only make cheap yarns because the unfavorable climate would not permit fine work. Result—poverty almost to slavery, unpainted shanties, no transportation, no paved streets, no amusements except white mule and the hook worm.

To-day, with Northern capital, added to the best of New England's occasional good air conditions in their plant, they make fine goods, the operatives are prosperous, the towns are modern and fit to live in. Many industries have been similarly revolutionized, and many more will be to great mutual benefit, all by the taking of ideal climate to the industry dependent upon it. All who are in the business of manufacturing weather, should feel proud that this vocation has more reward than just making a living.

What is weather—hot or cold, wet or dry, windy or still, clear or cloudy, each alternate is needed for some industry. All can be done

Paper presented before Philadelphia Chapter.

mechanically provided the rules of the game are known. The *psychrometric chart,* Fig. 1, shows them all, and for this reason I am taking the liberty of introducing this chart and asking that everyone get on better terms with it.

The terms used may not need any explanation, but in spite of this fact, I am going to try to express these from a little different point of view than is general at present.

First, there is a dry-bulb of the mercury or other liquid thermometer which is familiar and which, by the way, is of no use at all by itself. The standard is arbitrary and only a means of comparison. Heat, being molecular motion, the observed difference of dry-bulb temperatures only indicates the rate of flow from one condition to the other, just as the head or water pressure or voltage in electricity indicates the pressure available. There is nothing quantitative about it, and consequently the reading is comparatively useless alone.

Second, is the wet-bulb temperature which is quantitative and therefore extremely useful. This can be readily observed by placing a wet wicking on the ordinary dry-bulb thermometer with air circulated over it. This gives a depressed reading, proportional to the evaporation of the water from the wicking. As this temperature reading is the result of the sum of the latent and sensible heat in the air and vapor it indicates the total heat present. In fact, a wet thermometer could easily be calibrated in terms of total heat above zero or any other base. No calculations are possible without first obtaining a knowledge of the wet-bulb, and yet curiously general interest in this reading is comparatively recent.

Third, is relative humidity which everyone talks about and which occupies a position in all weather reports out of proportion to its real significance. It is only a ratio and is really an expression of the superheat present as in steam on a temperature plane more familiar to you all. Suppose the mixture of water vapor and air is saturated at 50 deg. and then superheated to 70 deg. in the absence of water. It is then one-half saturated or the relative humidity is 50 per cent.

Fourth, absolute humidity is the statement of the weight of water vapor present in the air and is usually expressed either in grains, per cu. ft. or per lb.of air.

Fifth, the dew-point is the saturation temperature. No more water vapor can be held in the space at the same pressure and temperature. If you remove heat it must condense just like steam always does when it is saturated and cooled.

Sixth, vapor pressure is the term usually treated with discreet silence, but without reason as there is no great mystery about it. The atmospheric pressure is made up of the sum of the vapor (steam) pressure and the column of the air which can be likened to a container, so there must necessarily be a

definite pressure for each saturation temperature just as there is in a boiler, with the modification that this boiler is elastic and explodes at the boiling point.

This is the vernacular of weather, and how all these points may be obtained from the chart and become useful will be shown.

The sling psychrometer, Fig. 2, is the little device with the big name which it deserves for it tells so much. The extended thermometer is covered with a wet wicking and then mounted on a handle so as to be swung readily. The average of several readings will give the most accurate results.

Suppose that the wet wick shows 71 deg. and the dry-bulb 90 deg. without reference to the chart. The drying characteristics of the air in which the instrument was slung are immediately known. A cubic foot of air is cooled approximately 8½ deg. for each grain of water evaporated and since the wet bulb is the limit of evaporative cooling, 19 deg. are available, which, when divided by 8½, leaves

FIG. 2. THE SLING PSYCHROMETER

for evaporation about 2¼ grains per cubic foot, if the drying devices were 100 per cent efficient as in the case of a spray type humidifier. Of course, the average dryer does not run much over 60 per cent efficient.

Using the chart, Fig. 3, all of the other characteristics of the surrounding air can be learned. First, by putting a pencil on the intersection of these two temperatures, namely, 71 deg. wet and 90 deg. dry, at (1) it is found that the air is 40 per cent saturated with water vapor.

To obtain the actual weight of water vapor present per lb. of air, go over from 90 deg. saturated at 2 to the column giving grains per lb. and it will be found that 216.4 grains at 3, the air being 40 per cent saturated, contains 40 per cent of this amount or 86.56 grains.

If it is desirable to know what the air weighs so that energies may be calculated, transferring the saturation point of the 90 deg. dry bulb at 6 to cu. ft. per lb., shows 14.54 cu. ft. per lb. at 7. Transferring the 90 deg. dry air at 4 to the calibration at 5, the result is 13.85, giving a difference of 0.685 cu. ft. As the air is 40 per cent saturated, 40 per cent of this difference or 0.274 added to 13.85 cu. ft. gives 14.12 cu. ft. per lb. under the conditions assumed. It is interesting to note that contrary to the average opinion, saturated air is lighter than dry air and that the moisture subtracts rather than adds to the weight of the mixture; hence the falling barometer as an indication of a coming storm.

If this air is to be dehumidified, the energy contained per pound would be important as it would be necessary to calculate the refrigeration required for subtracting the energy from the mixture. Transferring the wet-bulb temperature at 71 deg. saturated at 8 to the total heat above zero at 9 and over to the calibration column at the left gives 34.3 B.t.u. per lb. at 10.

To refrigerate this mixture to say 60 deg., 8.3 B.t.u. must be removed. So, dehumidifying becomes a simple proposition particularly when it is realized that the wet-bulb of the mixture indicates the total energy and could be so calibrated on the observation or recording instruments.

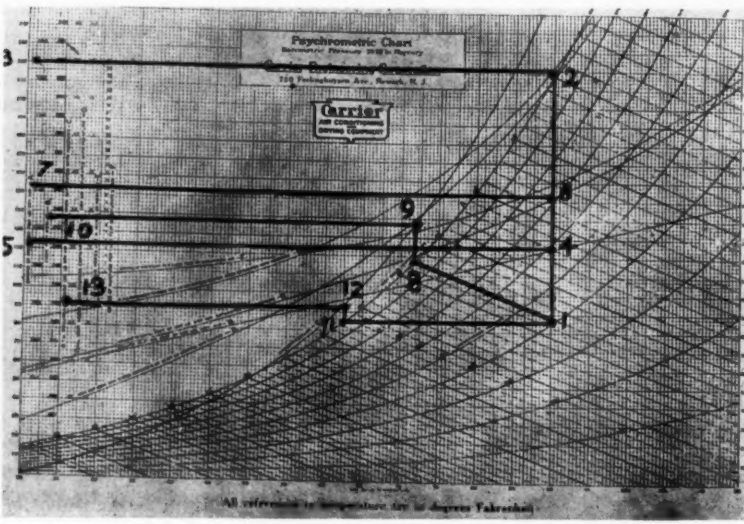

FIG. 3. CHART SHOWING PROCEDURE IN DETERMINING CHARACTERISTICS OF AIR

The vapor pressure is frequently important as mentioned before, and is solely the junction of the dew-point and bears no relation to either the dry or the wet-bulb. In this case, follow across to the dew-point and find 62½ deg. at 11. Thence up to the vapor pressure curve at 12 and over to the calibrated column which indicates 0.57 in. of mercury at 13.

Interest in this is due to the fact that if this air meets a hygroscopic material of a less vapor pressure, the material will gain weight from the equalizing of pressures and absorption of moisture and give off heat until equalization is reached. Or if the material is of a greater vapor pressure, it will lose weight by drying out and be cooled.

Knowing this factor it is easy to make material dry or regain moisture at will and is very useful in many industries, particularly

in textiles. From this it will be noted that the psychrometric chart is really useful in solving any kind of an air conditioning problem that may present itself, and is the key or alphabet, to the language of manufactured weather.

The operations of a humidifying system during both summer and winter in an industrial plant are shown graphically in Fig. 4. Assuming that silk is being fabricated and that it is desired to maintain 11 per cent regain in the material throughout the processes as it is best fabricated with this moisture content. Therefore 70 per cent humidity throughout the year is selected, as regain is more

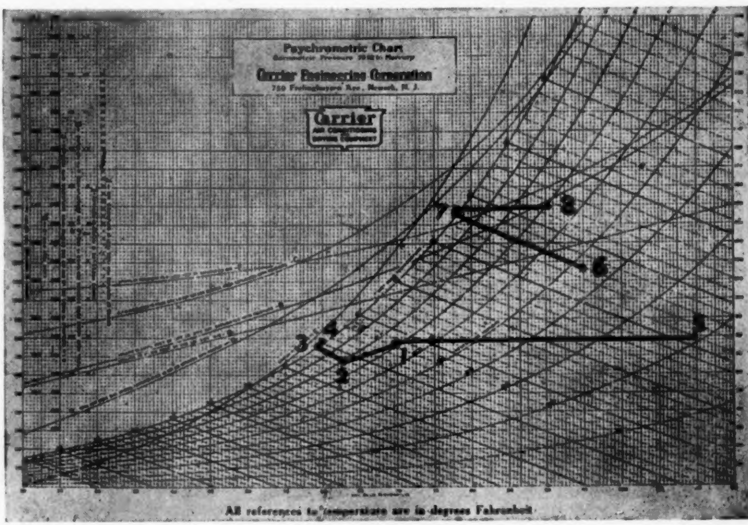

FIG. 4. CHART SHOWING OPERATIONS OF HUMIDIFYING SYSTEM IN AN INDUSTRIAL PLANT

nearly in accord with the relative humidity than any other single factor that is commercial to standardize.

Assume that it is zero outside, and that 1,000 lb. of air are required per minute to heat the building to 70 deg. and maintain 70 per cent humidity. Also assume that there is infiltration into this building of outside unconditioned air of 10 per cent the amount in circulation. The zero air contains only 5.7 grains per lb. while the 90 per cent return air contains 77 grains per lb. Therefore, when the 70 deg. air meets the 10 per cent zero air in the mixing chamber, by travelling along the point 1 to 2 it is found that the mixture has a 62½ deg. dry-bulb, 59 deg. wet-bulb, and 57 deg. dew-point.

It is an interesting fact that in this mixture, some condensation takes place in this assumed instance—the absolute humidity being

reduced from 77 grains per lb. to 69.3 per lb. or something over 1 lb. of moisture per minute which would indicate that drains are desirable in the fresh air intakes.

The air then comes in contact with the sprays and the instantaneous action is to cool the wet bulb to 59 or to the point of 3. It is then raised by the heat added to the water to the 60 deg. at 4, requiring in this case, the addition of 1800 heat units per min. where it arrives at the point of 4. Thence through the heaters to some point such as 5 in order to heat the room.

On entering the room, it is cooled by transmission from the building back to 70 deg. without either the addition or subtraction

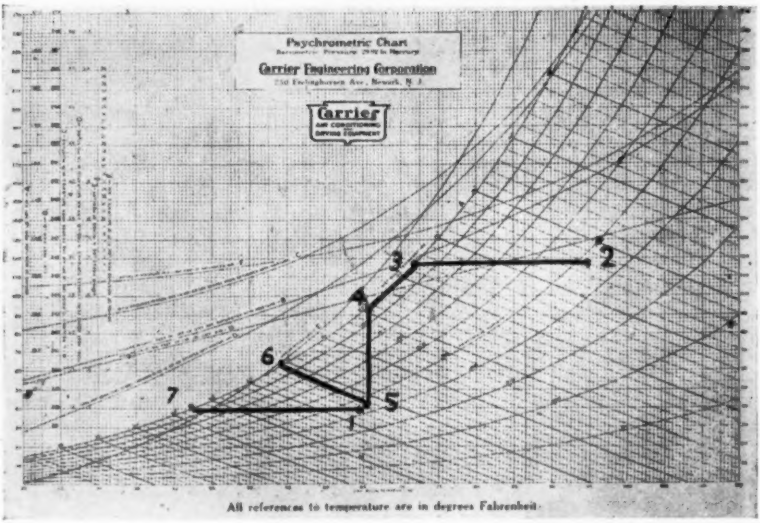

FIG. 5. CHART SHOWING METHOD OF WORKING OUT PROBLEM IN DEHUMIDIFYING CANDY FACTORY PACKING ROOM

of moisture and it arrives at the desired temperature and humidity at 70 deg. and 70 per cent.

In many mills moisture is absorbed by the stock or by infiltration, thus lowering the dew-point previously observed at the apparatus. This is usually compensated for by an auxiliary steam jet in the air distribution, after passing the heaters.

The summer cycle is very much simpler, as all fresh air is used. Assume that it is 95 deg. outside with a 78 deg. wet bulb, which is about the miximum in this vicinity. This would start at a point marked 6 where the air comes in contact with the water sprays and is cooled along the wet bulb line to 78 deg. at saturation, at point 7.

No energy is added or subtracted in this process but sensible heat

is converted into latent heat. The air enters the room at the point marked 7 in sufficient quantity to absorb all of the internal heat of the plant such as power, people, sunlight, radiation and transmission and rises to the point marked 8, or 89 deg., where it reaches the 70 per cent curve. The mill, in this case, is 6 deg. cooler than outdoors without any expense of operation beyond that of moving the air and pumping the water.

This phenomenon of evaporative cooling is probably one of the most valuable applications that have been made to the equipment of a textile mill since it makes possible and convenient the placing of

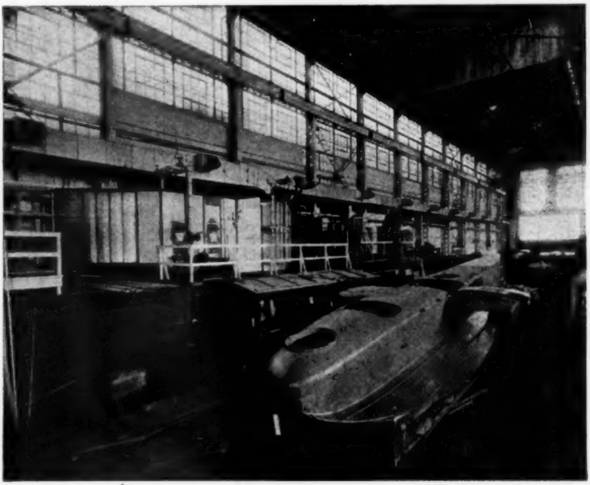

FIG. 6. TYPICAL HUMIDIFYING EQUIPMENT IN FINAL ASSEMBLY ROOM OF U. S. NAVAL AIRCRAFT FACTORY, PHILADELPHIA, PA.

a cotton mill in a cotton field and the production of fine yarns under livable conditions.

While the appearance of a humidifier and a dehumidifier are somewhat similar, their action does not depend upon the same laws. A humidifier is a cooling device only because of the evaporative characteristics of water. There is no change in energy or actual work done with the exception that some of the sensible heat of the air and water vapor mixture is converted into latent heat.

Water is in no way valuable in a dehumidifier except that it is an easily obtainable, cheap medium for presenting cold contact surface with the air to be cooled and the water vapor to be condensed. To follow this latter action on the chart, Fig. 5, assume that the problem in hand is a packing room in a candy factory with a room temperature of 65 deg. and 45 per cent relative humidity is desired, that a spray type of dehumidifier is to be used and that the day

outside is 95 deg. dry-bulb and 78 deg. wet-bulb. Further, that 1000 cf air per min. is required to do the work of which 95 per cent is in constant re-circulation, augmented by 5 per cent outside air at the apparatus for ventilation.

The air returns from the room at 65 deg. dry-bulb, and 52½ deg. wet-bulb, which has been designated on the chart as point 1. Five per cent fresh air enters at the point marked 2 in the fresh and return air mixing chamber, and is first cooled by contact with the 65 deg. air to point 3 or 71½ deg. This is on the saturation curve and by cooling to point 4, that is 66 deg., condensation takes place.

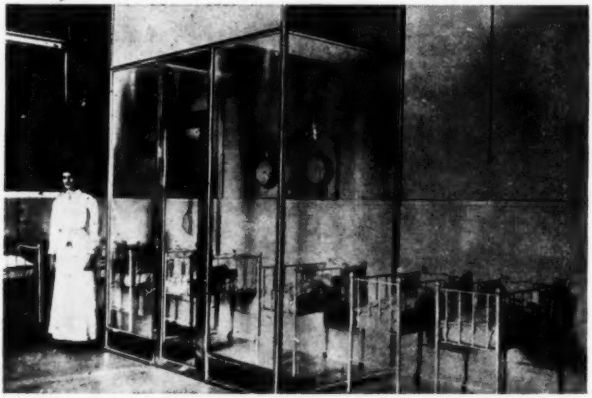

FIG. 7. AIR CONDITIONING EQUIPMENT ESSENTIAL PART OF BABY INCUBATOR

From point 4 to point 5, the relative humidity of this fresh air is further reduced by dilution in combining with the return air. In the meantime the return air has been raised from 65 deg. dry-bulb to 66 deg. approximately, and from 52½ deg. wet-bulb to 54 deg. wet-bulb approximately at 5.

At point 5, the air enters the dehumidifier and is probably instantaneously cooled by evaporation to point 6. No useful work having as yet been done. From point 5 to point 7 there is an actual absorption of energy reducing the air to its desired dew-point of 42 deg. Here not only is the temperature reduced but condensation takes place.

At point 7, the air enters the room and absorbs heat due to the process, power, infiltration, radiation, etc., and reaches its destination at the desired temperature of 65 deg. and 45 per cent humidity. The cycle is then repeated.

It is interesting to note that in the fresh air intake, in this assumed instance, 4½ tons of work are transferred from the return air to the fresh air, which will account for the usual difficulty in obtaining a thorough mixture. Also, that in this action, some 12¼ lb. of water

per hour are condensed which will readily show the necessity for water tight or insulated fresh air intakes for dehumidifiers.

All of the useful work was done between points 6 and 7 which by reference to the total heat curve before mentioned, would show that 6 1/10 B.t.u. had been removed per pound of air, or 30½ tons of refrigeration for the 1000 lb. of air.

INSURING SUCCESSFUL INSTALLATIONS

The idea of combining illustrations of this uninteresting chart with one or two pictures of actual every-day weather making ma-

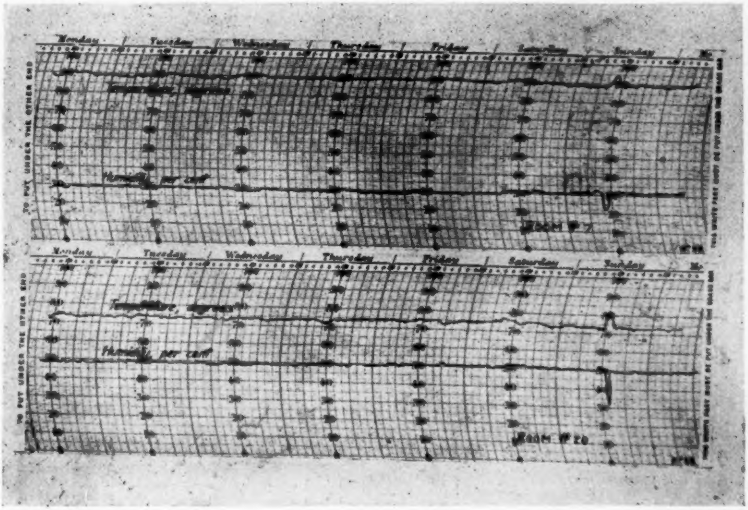

FIG. 8. CHART SHOWING HOW DIFFERENT TEMPERATURE AND HUMIDITY CONDITIONS ARE MAINTAINED SIMULTANEOUSLY IN DIFFERENT PARTS OF SAME PLANT

chines is two-fold: *first,* to show how thoroughly they are interlocked and that somebody must know the key and how to use it before a successful installation can be made; *second,* to show that air conditioning is no black art, but is founded on the knowledge of natural physical laws.

There is practically no limits to the size of any enclosure that may be controlled as to climate. Weather may be furnished either wholesale or retail. Fig. 6 showing one of the many units at the Naval Aircraft Factory at League Island is wholesale and made possible by wholesale production of aircraft. Fig. 7 being a baby incubator which saves many lives each year is retail. But, in both cases, the benefit to humanity brings joy to the work and makes the game of going nature one better worth while.

The manufacture of weather is dependable too, thoroughly commercial and economical. It is easy to give different conditions in various parts of an industrial plant with one central weather machine, for example note the two rooms Fig. 8 in which four different processes 90 deg. and 30 per cent, and 72 deg. and 50 per cent were produced simultaneously. Again it can be standardized whatever the outside weather vagaries.

Fig. 9 shows the relation between outside and inside humidities and proves at once that any hygroscopic material would vary in its characteristics if nature was not interfered with when she becomes wayward.

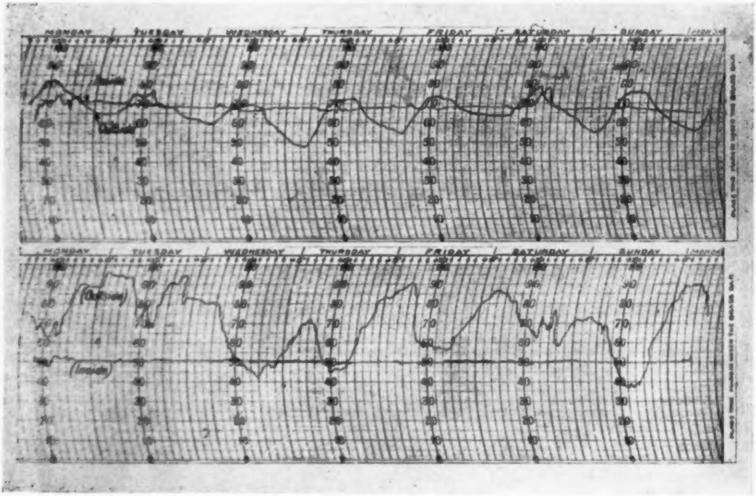

FIG. 9. CHART SHOWING RELATION BETWEEN INSIDE AND OUTSIDE HUMIDITIES

In Fig. 10 can be seen the dough room of a large baking company where conditioned air is employed to control the bacteriological process which is carried out in the raising of dough.

When it is considered that man-made-weather has been applied to literally hundreds of industries making some possible, improving others, reducing the cost of still more, it behooves all of us to take a keen interest in climate, minister to it when needed and above all learn the alphabet of its language and don't despise the psychrometric chart.

DISCUSSION

LEE NUSBAUM: I should like to ask whether the chart takes care of the introduction of heat from a motor and things of that kind?

THE AUTHOR: The air enters the building at the outside wet-bulb temperature. In the case I assumed it was 95 dry-bulb outside and the wet-bulb, 78. It was introduced to the building at 78 in sufficient quantity to absorb all the sources of heat, rising 11 deg. It is assumed that enough air is introduced to absorb heat from all sources when rising 11 deg. The reason I say 11 is that the difference between the dry-bulb and the dew-point must be 11 deg. to give 70 per cent. We take advantage of the fact that the air enters at the outside wet-bulb at 78 deg.

MR. NUSSBAUM: And you take care of it with the quantity of air?

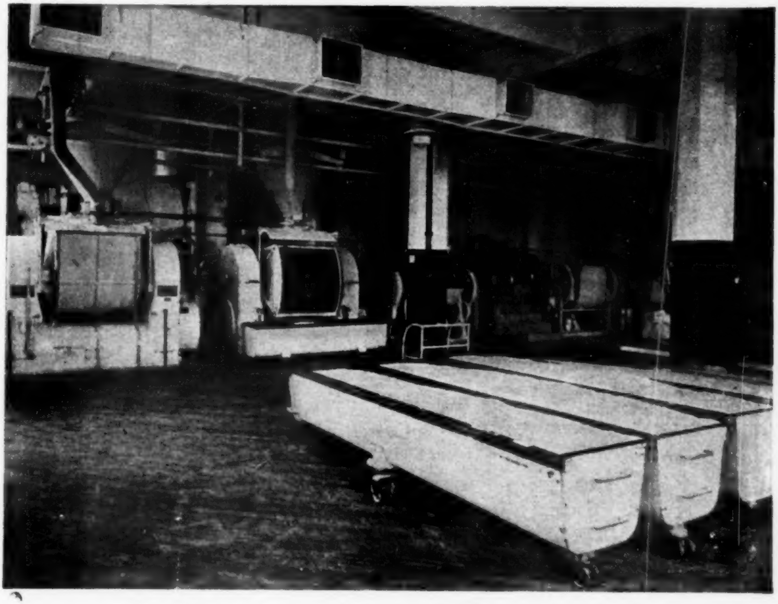

FIG. 10. BACTERIOLOGICAL PROCESS IN DOUGH ROOM OF BUFFALO BAKING PLANT CONTROLLED BY AIR CONDITIONING APPARATUS

THE AUTHOR: The quantity of air must be sufficient to absorb all the heat in 11 deg. rise to gain 70 per cent.

MR. NUSBAUM: You work to get a certain quantity of air to take care of that extra heat and then you can use from that the quantity of air you are going to handle.

THE AUTHOR: No. The starting point is the quantity of heat to be absorbed.

MR. NUSBAUM: How can you arrive at anything like that on your chart on an unknown quantity?

THE AUTHOR: You won't find the quantity of heat on the chart; it is in the mill and its determination is the first step in your calculation.

MR. NUSBAUM: How will you obtain on your chart the quantity of air required to take care of that extra heat introduced? You say you start at so many B.t.u. How do you know how many B.t.u. you absorb?

THE AUTHOR: We sum them up by making a survey of the building.

MR. NUSBAUM: There is nothing that you can get accurately on a thing like that, of course.

THE AUTHOR: That we are able to get it with fair accuracy is proven by uniformly successful results under many widely different conditions. This depends, of course, upon the engineering experience of the individual or individuals who make the survey.

No. 676

RECIRCULATION OF AIR IN VENTILATION

By Edwin S. Hallett[1], St. Louis, Mo.

Member

A DISCUSSION of matters pertaining to the recent progress in the handling of air in ventilation should be prefaced by a brief historical statement sketching the several steps which have been taken.

In 1917, when the writer began anew the study of the heating and ventilation of school buildings, he found that current literature reflected the present status of the subject. The American Society of Heating and Ventilating Engineers was becoming more active, and a number of able engineers were giving much time and thought to the subject of heating and ventilation. Apparently the only reliable knowledge of the subject was to be found in the advertising and private records of manufacturers of apparatus. With private laboratories not open to the professional engineer, the feeling that a real scientific laboratory should be created for supplying reliable data became so strong, about this time, that the Society took this bold step and established its Research Laboratory in the U. S. Bureau of Mines Experiment Station, Pittsburgh, Pa.

At the various meetings of the Society in New York City and elsewhere, reports have been received in large numbers to the effect that in most of the plants where both the fan system and direct radiation in the rooms were installed, the fans were shut down when the building could be heated without them. In the states where ventilation was compulsory, the fans were installed but the laws were not effective in keeping them in operation. The question was asked everywhere, "What is the use of wasting money on ventilation if they don't use it?" The world war coming on at that time doubled the cost of fuel and added another reason for sidestepping ventilation. Frequent remarks by interested persons were to the effect that "We would like to have ventilation if we could afford it but we think we cannot," and so the desire was not realized. The situation for ventilation, bad as it was, seemed fairly general. For an engineer who was out to actually see better school ventilation it was dire disappointment.

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Paper presented before the St. Louis Chapter.

In the St. Louis school buildings, another series of problems presented themselves. The school buildings erected in recent years there have been equipped with the all-blast plenum system which stood every test of theory or practice. There were no defects as far as fundamentals of ventilation were concerned and it required only the study of details of the mechanical features through collaboration with the architect in order to correct them in faulty design or construction.

While studying the conditions, tests were made of the air as to its qualities for ventilation as it came in at the usual intake windows and of air which was being discharged through the vent stacks on the roof. It was surprising to learn that the discharged used air was much more suitable for breathing than the air coming in through the "fresh air" intake. This situation may be explained by the fact that in the large cities there can be no such thing as clean pure air with the invisible presence of dust and fumes. These minute particles of contamination had to be removed by the air washer, and when the air was freed of the dust, soluble gases and fumes it had to be humidified and warmed to the correct room temperature so that it might move gently across the room and be discharged on the roof in almost as good condition as when it entered the room. Little oxygen was lost and a faint odor was picked up from the exhaled air.

The crux of the whole ventilation problem revolved about the discharged air, perfect except for the almost imperceptible human odor, but withal it was much superior to the air in the plain direct-radiator-heated schools. However, the occupants in the radiator rooms did not notice the odors, the sense of smell apparently becoming dulled. The fine odor-carrying dust fell into crevices, but still there was an ever-pervading odor present. School-houses with the maximum supply of air and no recirculation were particularly susceptible to these odors, as no fan system alone will remove them, while the recirculation of the air greatly increases them. At this juncture it was clear that some means must be found to get rid of the odors, and it was equally clear that all ventilation must hinge upon the success of recirculation. There was actually but one obstacle to it and that was the accumulation of odors.

It was apparent that the successful recirculation of the air would mean a complete revolution of heating and ventilating methods and a great modification in the design of apparatus to be used. It was necessary to find a means of restoring the air to normal purity without adding any kind of chemical or drug to the air. The idea from the beginning was to get back to nature's own fresh air as found at sea or in the uninhabited wastes of land. Naturally, ozone was suggested but the outlook for it was discouraging due to the adverse reports that had been made on its use by prominent experimentors. But the successful use of ozone in the London subway was uncontroverted. In the mountains and sparsely settled districts of the West abundant evidences of ozone in the natural air was an apparent contrast to its evident scarcity in cities. This fact seemed strange in that the concentration in nature at the highest has always been very low and thus it was assumed that some chemical affinity for the waste matter of the cities quickly consumed all ozone present.

Then came the new information from special research laboratories, principally that of Dr. F. O. Anderegg of the chemical laboratory, Purdue University. In pursuing a purely scientific investigation for a number of years Dr. Anderegg studied the corona discharge by which ozone is chiefly produced for ventilation purposes. He discovered that with the alternating current discharge against a dielectric that ozone was produced in small quantities at quite low voltages, even down to 500 volts, and the output increased very rapidly as the voltage increased. He also discovered that these rays produced nitrogen oxides, and that all previous experimentors had confused ozone with the mixture of ozone and nitrogen peroxide. The earlier ozone apparatus had used high voltages and the nitrogen content was relatively much higher. In other words the curve plotted for the nitrogen element was quite flat until the voltage passed the 5000 volt mark, when it quickly rose in its relative content as compared with that of ozone. This discovery was the key that unlocked the door for ozone in ventilation. By using a voltage under 5000 volts the nitrogen peroxide was so small as to be of no consequence. The Research Laboratory at Pittsburgh took up the matter and after many months of investigation with two machines which were offered for this work found the facts to be as stated above.

Another feature of the ozone problem was solved by the St. Louis experience. Ozone had everywhere been used in unknown but high concentrations. It seemed not to have occurred to most experimentors that ozone could be present without an odor. The theoretical critics of ozone always began by saying they did not like the smell of it. The odor and avoidance of an artificial air condition were enough to eliminate its use. However, the use of ozone similar to that of heat and all the rest of nature's resources must be used with judgment. The outcome was the development of a practical and safe means of controlling the output of ozone. The apparatus was designed in convenient units approximately as required for a school room. The apparatus was installed near the fan, either before or after, to secure perfect diffusion in the air. The concentration of ozone varied by the use of taps taken out of the transformer, or by a rheostat which used voltage between 3000 volts and 5000 volts, but in every case the control was locked or was inaccessible to the janitor or persons about the premises. The concentration when once adjusted by an expert did not have to be lowered. In fact, it was the custom to set it somewhat lower than normal at first installation and gradually to come to the correct amount. The correct concentration was just below that producing an ozone odor, approximately one part in two million of air.

The results of the use of ozone in the St. Louis schools have been described in many papers and have been seen by thousands of visitors. Its use has been a most complete success and has accomplished more than could have been anticipated. All the new schools designed in the last four years have had the ozone apparatus installed as part of standard heating equipment. In all about 43 large schools have been so equipped to date, and not a single instance of injury or suspected injury has oc-

curred. In fact the enthusiasm with which it has been received by teachers has been remarkable.

Some facts should, perhaps, be related at this point to show the effect which ozone in the schools has had on the health of the pupils and teachers, but suffice it to say that the children under normal weight at the date of installation of ozone have rapidly risen to normal weight. Undoubtedly there was a slight exhilaration or freedom from fatigue. So heating and ventilation engineers must take cognizance of the effect ozone has had upon the odors and effluvia of air used for ventilation.

The next step in advance was the modification of the building and heating apparatus to coincide with the new condition coincident to the recirculation of the air. It meant the recasting of every item of the design. At the outset, the new condition called for boilers of one-third of the former capacity. A 100 hp. standard tubular boiler was actually large for the standard 1,000,000 cu. ft. school building. The former practice called for 300 hp. The corridors and all rooms not filled continuously with pupils had to be supplied with warm air in quantities just sufficient to heat the space. With the air flow in class rooms adjusted to the velocity most pleasing to the teacher, no drafts were ever complained of, and no odors or other kind of stuffy air found present. In such a building the fan with a capacity of 45,000 cu. ft. was run at full speed so that the governor could maintain a constant speed on the fan. A larger fan was more difficult of nice control. This in effect reduced the fan size to one-third of former practice but it did not reduce the necessary and proper air motion in the school rooms. The air motion had to be maintained just short of what a teacher would call a draft.

Recirculation produced a new problem in the air washer. When recirculating 80 per cent or 90 per cent of the air with a high outside humidity the accumulated humidity of several returns of the air brought the humidity too high, if the standard mist nozzle type of washer were used. The air had to be re-passed through the washer as it removed dust which was carried in by the children. So the washer was designed on the carrier type but with the mist nozzles supplanted by a row of flooding nozzles placed at or over the top of the scrubber blades. It was demonstrated that the principal dust elimination was accomplished by the wet surfaces of the closely spaced blades. The purpose of the mist nozzle was to humidify. It has been found in the new schools so constructed that the dust and bacteria count has been extremely low.

The best results have come from buildings planned for recirculation of the air. The building including the heating plant was highly efficient and convenient. The general layout was made convenient for the heating apparatus so that the shortest runs were made of pipe and ducts. As an illustration a main central entrance was used and the fan and air washer were installed under the stairway and the ducts, while heating coils occupied the ground floor corridor. A thin shaft, full room width, came down on one or both sides of the stairway, from attic to basement for the return of the air from the attic to the fan. The branch ducts for the several rooms ran up in 12 in. slots across the ends of the rooms rather

than the larger rectangular flues, unless the old style wardrobe rooms were still used. The fact of supplying more than half of the building with warm air trunk ducts resulted in a great economy of space. It made provision for ample sizes in the room ducts. The plan provided an air pressure in the class rooms at all times. Corridor doors were kept closed with door checks. This pressure pushed the vented air on up to the attic duct and back to the fan. The fresh air intake connected with this return duct and only enough new air was taken in to replace the loses in the building, thus bringing about a minimum reduction. No tempering coils were required. The low pressure steam engine furnished the most economical power for the fan and air washer pump. The use of the exhaust steam for heating made this power less than that of any service company. The latest type of thermostatic control did away with rubber or other perishable material and proved extremely durable and sensitive, all dampers were operated by remote air pressure control.

Only in extremely moist weather has it been necessary to by-pass any of the returning air to keep down the humidity, provision being made for that if necessary. The cold air intake or the fresh air intake had to be left wide open at all times. It astonished visitors when told that the plant would recirculate 90 per cent of the air with the fresh air intake wide open at all times. The new air taken into the building was only what escaped through leakage and the toilet vents. The windows were weather stripped when new, and the frames were all plastic calked, and the leakage was very small in the new buildings. Formerly when all the air was wasted the leakage was not important but with every cubic foot of air carrying heat units it became a matter of cash value. Every avenue of escape of these B.t.u.'s was safely guarded. The open space in the flues or chases which carried the heat ducts to the several rooms was used to conduct the vented or returned air which was received in the attic by an insulated trunk duct, which carried this air to air shaft returning to the fan room. No metal duct was used for the air escaping from the room vents until it reached the attic. The air after leaving the room was treasured just the same as the coal pile, which it represented. The most direct means were used to accomplish every movement. The plant was built compactly and made convenient for the janitor or engineer.

The details of construction here recited will not be applicable to every case or to any other case perhaps, but they are suggestive to the enterprising designer. It will be seen that the cost of such a plant is greatly reduced. In fact it is cheaper to build, and build with the highest quality of apparatus, than to use a plain radiator system without ventilation. It will not cost more than one-half what the former heating plants cost. By an effort to cheapen they could be built for much less, but no skimping of sizes or quantity has ever been done. The factors of safety are, in fact, larger than ever. The quantity of vento coils is 50 per cent larger than theoretical needs to enable the janitor to heat the building quickly.

A right state of mind must be developed among teachers with reference to ventilation, not for the sake of the ventilation system or the engineer, but for their own sakes. It ought to be apparent to them that the washed

and purified air which has passed quickly across a classroom is still purer and more wholesome than that which the wind has swept up through a filthy alley or over the refuse of a slaughter house. It will greatly add to their comfort to know that the air has been scrubbed and bleached by ozone until it is pure as the fresh laundered handkerchief. In contrast with this is the shower of dust and bad odors that fly in at the open window. The present highly perfected ventilation plants should be used as an educational feature in every school as a means of securing the full value from them. Nothing in a school has ever been done that ought to appeal to the imagination in a helpful way, so much as the new ventilation. Anybody can take a deep breath and feel fine out in Yellowstone Park, and the air of our schools is identically the same. Some enthusiasm in "sensing" that fact will bring forth equally good results in other cases.

Recirculation and repurification have been fully established in both scientific and practical methods. The very large applications in St. Louis have been demonstrated beyond any possible denial as the best air condition ever produced by any method. The economy in plant costs and in fuel costs are great but these should be forgotten when the health of the school is concerned. It has been gratifying beyond measure to have authentic confirmation of the fact of the rapid and general increase of weight to normal of the anaemic or pale children. The near future must reveal other advantages not here mentioned.

No. 677

STANDARD CODE FOR THE TESTING OF CENTRIFUGAL AND DISC FANS

IN order to establish a standard method of testing centrifugal and disc fans, and to provide a uniform basis for compiling fan performance data, this code has been framed by a joint committee from the *National Association of Fan Manufacturers* and the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS appointed for such purpose.

The Committee has endeavored to embody in this code methods that are technically correct and also practicable, in that provisions can be readily complied with from the standpoints of economy and convenience of application.

Throughout this code the following nomenclature will be used:

1. *Standard Air* is air weighing 0.07488 lb. per cu. ft. This weight corresponds to air having a barometric pressure of 29.92 in. of mercury, a dry-bulb temperature of 68 deg. fahr., and 50 per cent relative humidity.
2. *Static Pressure* (*SP*) is measured at right angles to the direction of flow.
3. *Total Head or Total Pressure* (*TP*) is measured by an impact tube.
4. *Velocity Head or Velocity Pressure* (*VP*) is the difference between the total pressure and the static pressure.
5. *Capacity* (*CFM*) is the cubic feet of air per minute handled by the fan.
6. *Horsepower Output* (*AHP*) of a fan, or air horsepower, is expressed by the formula:

$$AHP = \frac{CFM \times TP}{6356} \tag{1}$$

where CFM = capacity in cu. ft. per min.

TP = total pressure in in. of water.

(For derivation of Formula (1) see appendix.)

Code adopted at the Semi-Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Chicago, Ill., May, 1923.

7. *Horsepower Input* (*HP*) is the horsepower required to drive the fan.
8. *Mechanical Efficiency* (*ME*) of a fan is the ratio of horsepower output to horsepower input.
9. *Static Efficiency* (*SE*) is the mechanical efficiency multiplied by the ratio of static to total pressure.
10. *Fan Performance* is a statement of the capacity, pressure or pressures, speed, and horsepower input.
11. *Fan Characteristic* is a graphical presentation of fan performances throughout the full range from free delivery to no delivery at constant speed.

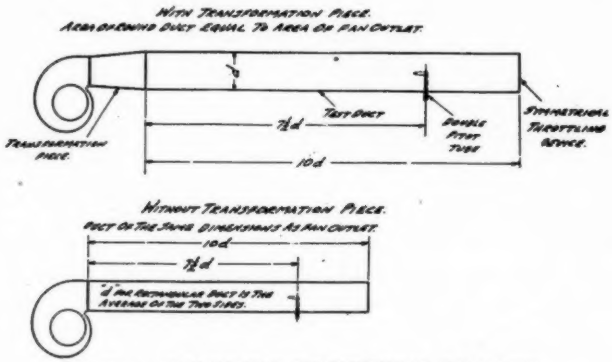


FIG. 1. SET-UP FOR CENTRIFUGAL FAN TEST

12. *Fan Discharge or Outlet* is the place provided for receiving a duct through which air leaves the fan.
13. *Fan Inlet* is the place provided for receiving a duct through which air enters the fan.

CENTRIFUGAL FAN TESTS

Test Duct: All centrifugal fans shall be tested with a discharge duct having an area equal to the fan discharge. This duct shall be straight, of uniform area and section and not less than 10 diameters in length. Readings shall be taken over a plane at right angles to and the pitot tube shall be parallel to the axis of the duct, with the impact end pointed upstream. The place where these readings are taken shall be three-quarters of the length of the test duct from the fan end—Fig. 1. The outer end of the test duct shall be equipped with an adjustable device for symmetrically throttling the air.

Number of Readings—Round Ducts: In all round ducts, not less than 20 readings shall be taken (not less than 2 traverses of 10 readings each along diameters at right angles to each other—Fig. 2) at the centers of areas of five equal area concentric zones across the section of the duct. In making tests with small ducts the calibrated-nozzle method of Sanford A. Moss may be used for determining volumes, Fig. 6. This method is fully described in the 1916 *Transactions* of the *American Society of Mechanical Engineers*.) Several nozzles with different size outlets are required to give the range of capacities. By making the fan outlet duct 20 diameters long and

introducing a throttling device midway the number of nozzles required may be reduced.

Number of Readings—Rectangular Ducts: In all rectangular ducts, the readings shall be taken in the center of equal areas over the cross section of the duct—Fig. 2. The number of equal spaces shall not be less than 16, and need not be more than 64 in any case. When less than 64 readings are taken, the number of equal spaces shall be such that the centers of these rectangular areas are not more than 6 in. apart.

Angle of Transformation Piece: In any transformation piece the greatest angle between the axis of the test duct and any longitudinal element shall not exceed 7 deg.

Pressure-Measuring Instruments: The standard instruments for measuring pressures shall be the double pitot tube and the manometer.

The pitot tube shall be the small orifice tube having static orifices not exceeding 2/100 in. (0.02 in.) in diameter. There shall be not less than 4 orifices, no orifice to be located at less than 8 tube diameters from the upstream end of the tube and an equal distance from the elbow—Fig. 3.

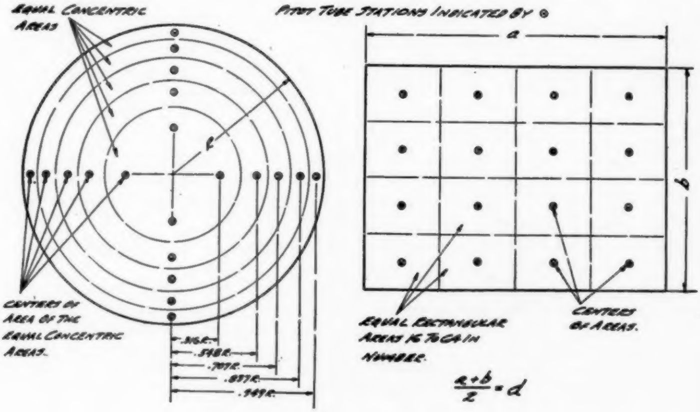


FIG. 2. DUCT TRAVERSES

The manometers or gauges for measuring static pressure and velocity pressure are to be filled with a light liquid-kerosene, gasoline, or alcohol, and calibrated in position by comparison with a water-filled hook gauge, the screw of which is also calibrated to a degree of accuracy commensurate with the accuracy of the proposed readings. This calibration shall be made with the rubber tubing attached, and the rubber tubing shall not be removed from the manometer after calibration. All joints and connections shall be air-tight.

Calibration of Instruments: All instruments shall be calibrated immediately before and after the test.

The manometer for measuring total pressure shall be vertical and water-filled—See Figs. 3 and 4.

Dynamometer: The cradle dynamometer is recommended for all power measurements. Direct connected motors carefully calibrated may be used. Where indirect drive is used friction tests shall be made to determine losses for each of the conditions of the tests.

Inlet or Outlet Obstructions: Fan performances are based on unobstructed inlets and outlets except for the structural features of the fan involved in bearings and bearing supports. If a fan is provided with bearings it shall be tested on its own shaft and in its own bearings, and shall be run for several hours to wear in. Fans with bearings or bearing supports in the inlets shall be so mounted for tests.

Number of Determinations: Eight or more determinations covering a range of performance from approximately free delivery to no delivery shall be made with approximately equal increment of capacity or discharge area.

Observations and Calculations: Simultaneous readings of total pressure, static pressure, velocity pressure, horsepower input, and speed shall be taken. Barometric pressure and wet- and dry-bulb temperatures shall be taken during each determination. Temperatures shall be taken both in the room and within the discharge duct as close as practicable to the point of traverse. Speed shall be maintained as nearly constant as possible throughout the entire test. Where tachometers are used they shall always be checked by revolution counters, readings being taken over a period of not less than one minute.

All air readings shall be corrected to standard air conditions at fan inlet, *i.e.*, air weighing 0.07488 lb. per cu. ft. These corrections shall be made based on the following relations: the pressures and the horsepower input vary directly as the weight of air at constant speed. The velocities corresponding to the velocity pressure readings shall be averaged to obtain the average velocity for each traverse. Average velocity pressures shall be calculated from the average velocities thus determined.

The velocity is determined from the velocity pressure by use of the following formula:

$$V = 1096.2 \frac{VP}{w} \qquad (2)$$

where V = velocity in ft. per min.
VP = velocity pressure in in. of water
w = weight in pounds per cu. ft. of the air in the duct

The values of w shall be taken from the tables of the United States Weather Bureau. (For derivation of Formula (2) see appendix.)

The capacity for each determination is the product of the average velocity and the duct area corrected for inlet conditions.

Capacity, static pressure, total pressure and horsepower input shall be corrected to constant speed. For speed corrections capacity varies directly as the speed, pressures as the square of the speed, and horsepower as the speed cubed for constant discharge conditions. The total pressure and the static pressure shall also be corrected for duct friction to give conditions at fan discharge by adding the loss of head due to friction of test duct between the fan outlet and pitot tube. This loss will be:

FOR ROUND DUCTS.

$$f = 0.0257 \frac{L}{D} VP \qquad (3)$$

where f = pressure loss in in. of water
VP = average velocity pressure in in. of water
L = distance from fan outlet to pitot tube in ft.
D = diameter of discharge duct in ft.

FOR RECTANGULAR DUCTS.

$$f = 0.01285\, L \frac{a + b}{ab} VP \qquad (4)$$

where a = long side of duct in ft.
b = short side of duct in ft.

(For discussion of friction Formulae (3) and (4) see appendix.)

Results: After the performance has been calculated for each determination the results, as corrected for standard air conditions and constant speed, shall be plotted using capacity as abscissæ, and pressures and horsepowers as ordinates. Smooth curves are then drawn through these points giving pressure and horsepower characteristics. From these data the mechanical efficiency is calculated.

The mechanical efficiency curve is then plotted against capacity. These curves form the basis from which fan performance tables and characteristic charts are computed.

In calculating the performance of fans of the same design and similar proportions, but of other sizes and at other speeds, the following law applies; up to and within 1½ lb. per sq. in. pressure difference, at the same peripheral speed and discharge condition the delivery varies as the square of the diameter of the wheel, or for different speeds the delivery varies as the cube of the diameter times the number of revolutions.

Exhausting Fans: In testing fans for exhaust pipe systems the capacity shall be determined by readings taken in the discharge duct in the manner already prescribed. To determine the total pressure there shall be provided an inlet duct having an area equal to the fan inlet and 6 diameters in length—Fig. 4. The total head or total pressure is the difference between the average absolute total pressure in the discharge duct and the average absolute total pressure in the inlet duct with additions to cover friction of inlet and outlet ducts between the points of measurement and the fan in accordance with the method previously described. The average absolute total pressure in the inlet duct shall be determined by adding to the absolute static

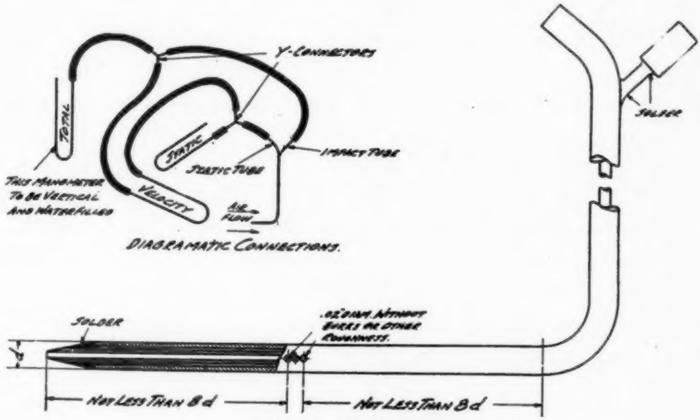


FIG. 3. PITOT TUBE

pressure in the inlet duct the calculated average velocity pressure in the inlet duct. (This total pressure in the inlet duct may also be obtained from the average traverse impact tube readings or more simply from the readings of impact tubes set up as shown in Fig. 4.) Readings in the inlet duct shall be taken at a place three-quarters of the inlet duct length from the entering end. The static pressure developed by the fan is the difference between the total head thus obtained and the velocity head in the discharge duct.

Tables based on such tests shall be clearly marked as for exhaust systems only and shall not be used for other purposes.

Tolerances: As there are errors of observation, instruments, and manufacture, it is recognized that there may be slight errors in the determinations of capacity, pressure, or horsepower. A variation of 2½ per cent in one of the above, *i.e.*, a tolerance factor of 1.025 applied to the mechanical efficiency, will not be considered excessive.

DISC AND PROPELLER FAN TESTS

This code on the testing of disc and propeller fans is framed on the basis of fans exhausting from a chamber, and all air readings must be taken on the inlet side. As in the case of centrifugal fans, so with disc and propeller fans, the standard instruments for measuring pressures shall be the double pitot tube and the manometer.

Test Set-Up: The test set-up for disc and propeller fans shall consist of an air-tight chamber of rectangular or drum form and of minimum size in proportion to the fan to be tested as shown in Fig. 5.

At the entrance side of the chamber shall be a duct 10 duct diameters in length, which length includes a transformation piece connecting to the chamber. The transformation piece shall be carried to the full dimensions of the end of the chamber and provided with an effective diffuser.

By restriction at the inlet end of the test duct the volumes handled shall be varied, and free delivery conditions shall be procured by the use of an auxiliary blower for obtaining zero or above static pressures in the chamber.

The capacity of the fan shall be determined by readings taken in the test ducts at the place indicated and in the manner already prescribed for centrifugal fan tests. The static pressure shall be determined by a reading of an impact tube in the large chamber.

Tables based on above test shall be designated for wall installation without ducts.

Disc Fans Tested Blowing: When disc fans are used to blow into a pipe in actual service the performance is different from the performance when mounted in a wall, and consequently allowance should be made for the modified condition. Disc fans for blowing purposes are to be tested with the same standard arrangement of apparatus as shown on Fig. 5, but in addition there must be placed on the discharge side of the fan a collar of the same diameter as the fan opening and 1 diameter long.

Tables based on such tests shall be designated for installation with ducts.

Mounting: Any obstruction to the flow of air such as a driving motor, pulley, bearings, framing, or the like, that forms part of the unit, whether normally on either the discharge or inlet side of the fan, shall be in its designed position so as to effect the test results to show the true performance of the fan as used in that unit.

APPENDIX

Derivation of Formula 1.

$$\text{Horse Power} = \frac{\text{Foot lb. per min.}}{33000}$$

$$= \frac{\text{Weight of Air per min.} \times H}{33000} = \frac{CFM \times w \times H}{33000}$$

Where w = air density

H = head in ft. of air

$$H = \frac{TP}{12} \times \frac{62.3}{w}$$

Where 62.3 = density of water

And TP is in in. of water

$$\text{Horse Power Output} = \frac{CFM \times w \times TP \times 62.3}{33000 \times 12 \times w}$$

$$AHP = \frac{CFM \times TP}{6356}$$

Derivation of Formula 2.

From the law of falling bodies

$$v = \sqrt{2gH}$$

Where v = velocity in ft. per sec.

g = acceleration due to gravity

H = head in ft.

Substituting for $$H = \frac{VP}{12} \times \frac{62.3}{w}$$

Where w = air density

We have in ft. per min. $$V = 60\,v = 60\sqrt{\frac{2g\,VP \times 62.3}{12 \times w}} = 1096.2\sqrt{\frac{VP}{w}}$$

Discussion of Formulae 3 *and* 4: The friction formulae approved in this code are the ones adopted by the U. S. Navy. Corrections for duct friction are a small percentage in fan tests and the Navy formulae have been found closely to approximate test results.

These formulae are based on the hydraulic formula:

$$\text{Loss of head due to friction} = K\frac{P}{A}lv^2$$

Where K = a constant from experiment
P = perimeter of duct
A = area of duct
l = length of duct in ft.
v = velocity in ft. per sec.

K in the Navy formula is 0.0001

When proper substitutions are made, Formulae 3 and 4 result.

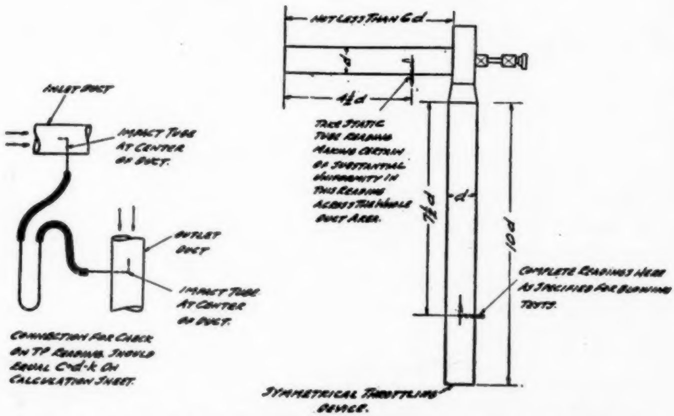


FIG. 4. SET-UP FOR TESTING CENTRIFUGAL EXHAUST FANS

Calculation Detail: In these calculations small letters have only the significance specifically given herein and do not refer in any way to any conventional meaning with which they may be associated in usual formulation.

In determining air densities inside ducts or chambers under suction, temperatures inside those chambers must be used and barometric pressures equal to the atmospheric barometer diminished by the static tube reading in that suction space in inches of mercury. The densities are taken from the Weather Bureau tables, corresponding to these conditions.

In the case of density, at fan discharge, for propeller fan calculations, the temperatures must be taken in the discharge from the fan and the barometric pressure directly from the reading of the barometer in the room or space into which the fan discharges.

Densities in discharge or pressure ducts are similarly determined from temperatures in the duct, the barometric pressure being the atmospheric pressure increased by the reading of the static tube in inches of mercury.

In most low-pressure fan testing the effect of the differences in the densities at various locations with respect to the fan is entirely negligible; but at some of the higher pressures in fan work such is not the case, and the various densities must be considered.

Calculation Sheet for Centrifugal Fans.

Blowing (see Fig. 1.)

a = average VP in discharge duct in in. of water
b = static pressure in discharge duct in inches of water
c = total pressure in discharge duct $= a + b$
d = revolutions per min.
e = density of air in discharge duct in lb. per cu. ft.
f = density of air at fan inlet in lb. per cu. ft.
y = area of discharge duct in sq. ft.
z = friction correction according to Formula (3) or (4)
$i = SP = b + z$
$j = TP = c + z$
k = horse power input HP

$$l = CFM = 1096.2\, y \sqrt{\frac{a}{e}} \times \frac{e}{f}$$

$$m = AHP = \frac{l \times j}{6356}$$

$$n = ME = \frac{m}{k}$$

$$o = SE = n\frac{i}{j}$$

In reducing to standard air multiply the TP, SP, and HP by the ratio $\frac{0.07488}{f}$.

Calculation Sheet for Centrifugal Fans.

Exhausting (See Fig. 4.)

a = average VP in discharge duct in in. of water
b = static pressure in discharge duct in in. of water
c = total pressure in discharge duct $= a + b$
d = static tube reading in inlet duct in in. of water.
e = revolutions per min.
f = density of air in discharge duct in lb. per cu. ft.
y = density of air at fan inlet in lb. per cu. ft.
z = area of discharge duct in sq. ft.
i = area of inlet duct in sq. ft.

$$j = CFM = 1096.2\, z \sqrt{\frac{a}{f}} \times \frac{f}{y}$$

$$k = \text{velocity pressure in inlet in in. of water} = \frac{y}{1.2}\left(\frac{f}{1000i}\right)^2$$

l = friction correction according to Formulae (3) and (4)
$m = TP = c + d - k + l$
$n = SP = m - a$
o = horse power input HP

$$p = AHP = \frac{j \times m}{\quad} \quad FCEF.$$

$$q = ME = \frac{p}{o}\,.$$

$$r = SE = q\,\frac{n}{m}$$

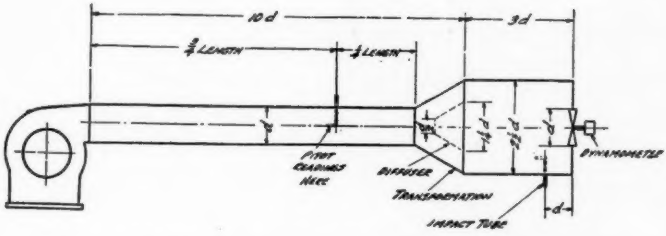
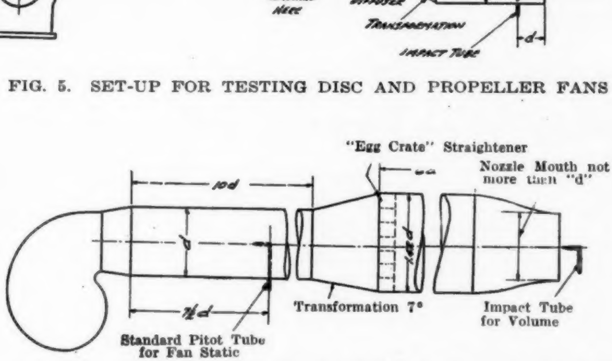


FIG. 5. SET-UP FOR TESTING DISC AND PROPELLER FANS



NOTE:—If the moss nozzle is more than 50% of the duct area, the nozzle must be reversed and the testing is no simpler than the pipe traverse with the enlarged section as shown, 1.42*d*. The nozzle can equal *d* one reading at the center taken and a coefficient of 0.98 used, thus taking advantage of the simplification of this method even at wide open.

FIG. 6. SET-UP FOR MOSS NOZZLE TEST WITH IMPACT TUBE

For the purposes of the calculations outlined all quantities are to be used with a positive sign understood and the various items evaluated according to the arithmetical processes indicated. There is apt to be some confusion about "*d*" which is, as read during the test, a negative quantity. The calculations have been arranged so that no attention is to be paid to this negative sign and the quantity used as a positive amount.

In reducing to standard air multiply *TP*, *SP*, and *HP* by the ratio $\dfrac{0.07488}{y}$.

Calculation Sheet for Propeller Fans.

(See Fig. 5.)

a = average velocity pressure in suction duct in in. of water
b = SP = reading of impact tube in large chamber in in. of water
c = area of suction duct in sq. ft.
d = area of circle of maximum fan-blade diameter; unless thimble is provided, in which case d = area of inside of thimble. If test is for blowing performance d = area of duct on fan outlet
e = air density in suction duct in lb. per cu. ft.
f = air density in large chamber in lb. per cu. ft.
y = air density at fan discharge in lb. per cu. ft.

$$z = CFM = 1096.2c\sqrt{\frac{a}{e}} \times \frac{e}{f}$$

$$i = z\frac{f}{y}$$

$$j = VP = \frac{y}{1.2}\left(\frac{i}{1000d}\right)$$

$$k = TP = b + j$$

$l =$ horse power input HP

$$m = AHP = \frac{z \times k}{6356}$$

$$n = ME = \frac{m}{l}$$

$$o = SE = n\frac{b}{k}$$

In reducing to standard air multiply the TP, SP, and HP by the ratio $\frac{0.07488}{f}$.

In Memoriam

Names	*Joined the Society*	*Died*
J. J. Gillett	May, 1921	January, 1923
A. H. Ross	March, 1920	February, 1923
H. L. Williams	June, 1911	February, 1923
J. B. Blake	June, 1910	July, 1923
B. W. Brady	June, 1919	August, 1923
W. J. Haynes	January, 1911	December, 1923
H. C. Mallory	July, 1903	December, 1923
W. W. Towson, Jr.	August, 1918	December, 1923

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